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FINAL REPORT

PRELIMINARY DESIGN OF A MINI-BRAYTON COMPRESSOR-ALTERNATOR-TURBINE (CAT) CONTRACT NAS3-16739

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APS-5440-R

MARCH 12, 1973

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COMPRESSOR-ALTERNATOR-TURBINE
(CAT)
CONTRACT NAS3-16739

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Power Project



AIRESEARCH MANUFACTURING COMPANY OF ARIZONA
A DIVISION OF THE GARRETT CORPORATION

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(CAT)
CONTRACT NAS3-16739

1. INTRODUCTION AND SUMMARY

This report, prepared by AiResearch Manufacturing Company of Arizona, A Division of The Garrett Corporation, is presented in fulfillment of the data requirements of the Mini-Brayton Contract NAS3-16739. This program consists of the following four tasks pertaining to the study of a 0.5 to 2.0 kw_e closed Brayton cycle power system.

Task I - Parametric Study
Task II - Off-Design Study
Task III - Preliminary Design
Task IV - Reporting

Many space missions encompassing near earth orbit, interplanetary and deep space vehicles are planned for the next twenty years. These missions will require an electrical power source which can supply from several hundred watts to a few thousand watts of power. Among the leading candidate power systems are isotope-Brayton, reactor thermoelectric (RTG) and solar-plus storage battery systems.

In the required power range, isotopic fuels offer the most promise as a heat source from which to make electricity. Since isotopes are costly, it follows that the efficiency of the power conversion system must be high. Of the three systems mentioned, the Brayton system has the best potential for high efficiency.



Missions of many years duration require a power conversion system of unprecedented reliability. The system must be able to operate without human attendance, must supply uninterrupted power, and must operate in the harsh environment of outer space. As simple a system design as possible with few moving parts and, if possible, no rubbing parts would be desirable if this high reliability is to be attained.

An extensive study was performed during 1972 by the NASA-Lewis Research Center to evaluate the isotope-Brayton electric power system. This system was compared, for several typical missions, on indices such as cost effectiveness, reliability, and performance, with other power system approaches.

The results of the study were reported in NASA TM X-68072 and presented at the Seventh IECEC in San Diego, California, on September 25-29, 1972. It was concluded that the cost effectiveness of the Brayton system depends upon the nature of the mission. For a number of unmanned satellite missions requiring from 600 to 3000 watts electric, the isotope-Brayton system cost, on a "dollars-per-watt" basis, was comparable to solar array/battery systems and about one-fourth that of RTG systems. These favorable economic considerations, combined with a 10 year Brayton technology base established at NASA-Lewis, stimulated further investigation of the mini-Brayton concept.

1.1 General Program Description

The purpose of this mini-Brayton study was to further develop design philosophy for low power level isotope-Brayton systems and to conduct a preliminary design for the rotating component of the system. The study has stressed, during Tasks I and II, the following design criteria:

- (a) Simplicity of design
- (b) High reliability



- (c) Long life (five to ten years)
- (d) High efficiency at low power level
- (e) Minimum extrapolation of proven technology
- (f) System versatility

Also an important goal for the design presented herein was mission versatility. It is desirable, particularly from a cost standpoint, to select one system that may be easily adapted to various mission power requirements.

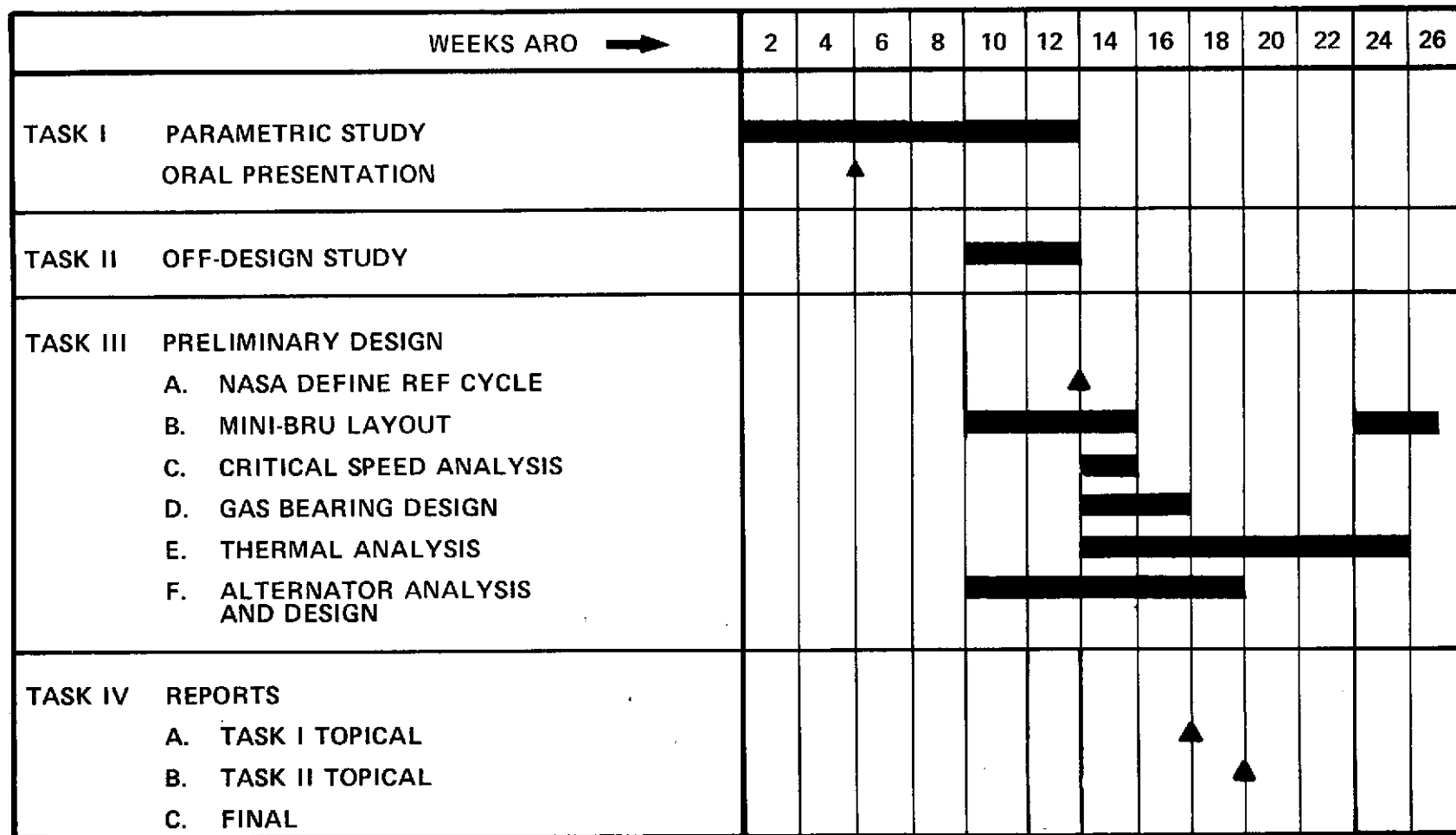
The schedule for the study program is shown in Figure 1-1. The contract start date was June 26, 1972. The results of the study have, to date, been very encouraging and some important design philosophy relative to very small Brayton systems has been postulated, explored, and defined. The reference cycle and system components are described in Section 1.3 of this report.

The balance of the study, Task III, was devoted to a preliminary design of the mini-Brayton (a schematic is presented in Figure 1-2 and a cross section is shown in Figure 1-3). The unique design features incorporated into the design include:

- (a) Hydrodynamic (self-pressurized) foil gas bearings
- (b) Gas cooling of the alternator and bearings (liquid loops not required anywhere in the power system--a direct or gas flow radiator is used)
- (c) A solid rotor Rice alternator
- (d) Motor start up of the CRU (combined rotating unit)



MINI-BRAYTON STUDY PROGRAM SCHEDULE



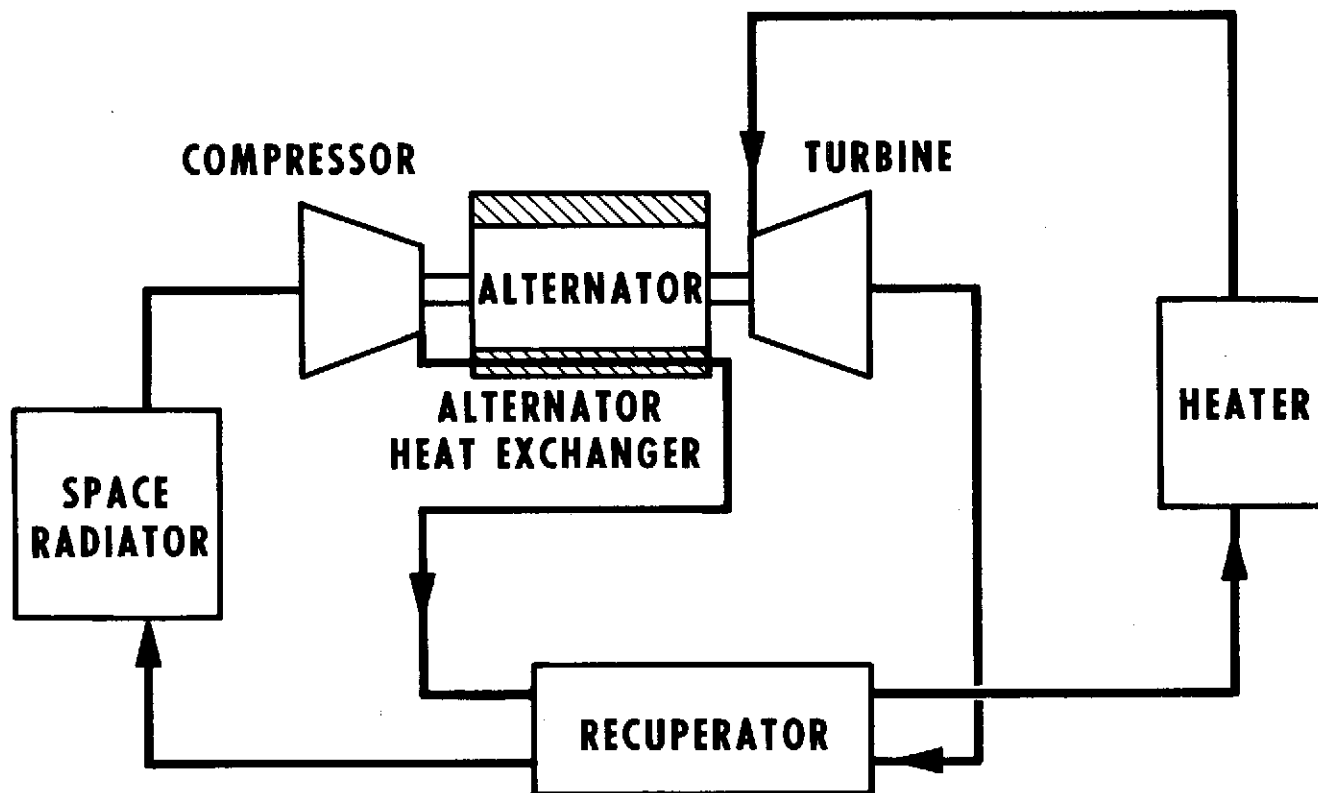
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TASK I. START DATE JUNE 26, 1972

FIGURE 1-1



MINI-BRU CYCLE SCHEMATIC



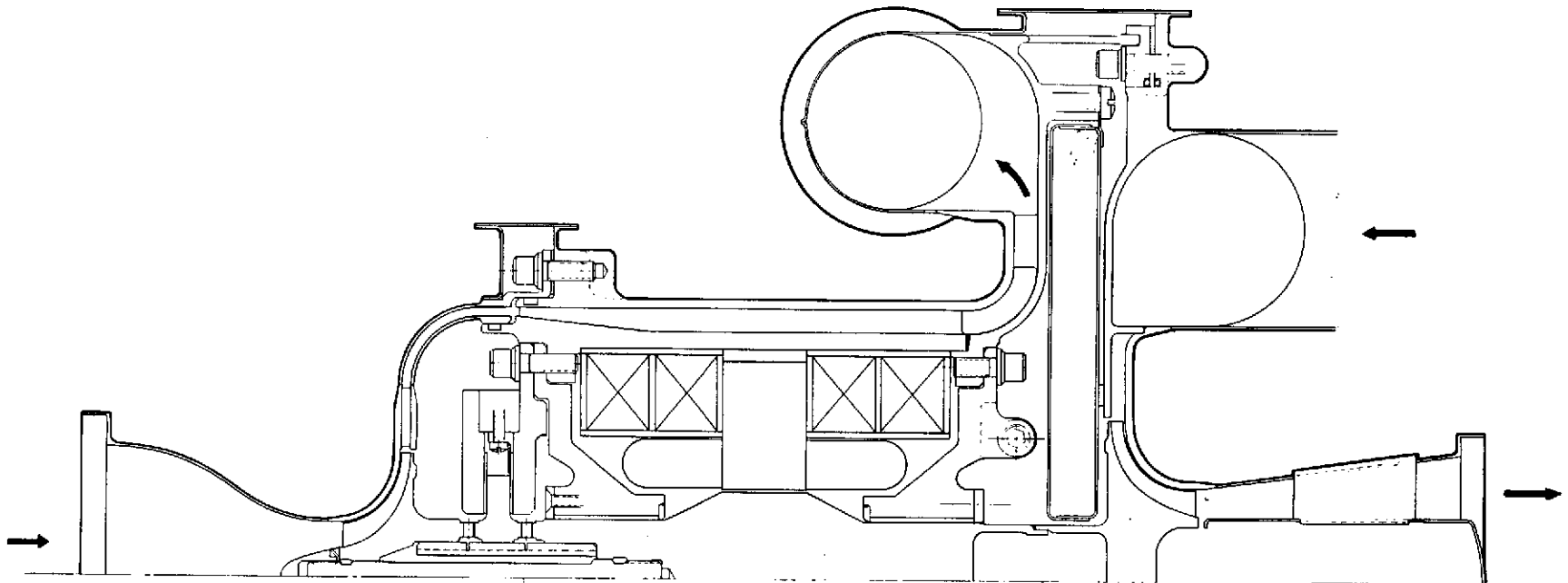
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FIGURE 1-2



MINI-BRAYTON ENGINE

0.5 TO 2.0 KWe



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FIGURE 1-3



Further discussion of the study objectives and guide lines is included in Section 1.2.

1.2 Program Objectives

Table 1-1 lists the program design goals. Optimum system characteristics over the entire power output range were determined by performing a wide-range parametric study. Mathematical models, in the form of a series of digital computer programs are used during the optimization procedure. These models, whose accuracy has been proven through correlation with test data for components and also for complete systems, provide the capability to explore design questions in great scope and depth in a short period of time.

A further important objective of Task I was, having defined component design characteristics, to assess the ability to develop those components to the degree necessary within the limitations of present technology. This is done by comparing the design requirements to those already proven with existing hardware.

Task II was performed using a sophisticated off-design performance computer program. With this program, the off-design performance of a fixed hardware system (i.e. a particular and unique system design) can be predicted. This program has been verified by comparison to actual systems in operation. Important conclusions were drawn from the Task II study relative to the number and type of systems required to cover the power spectrum of interest.

An additional important facet of the study was the determination of system performance sensitivity to the various individual design parameters. This question was studied in two parts. The first was the impact on system optimization of constraints placed on component performance. The second was the evaluation of the impact on system performance and/or weight of a component unable to meet its design goal (assuming this was discovered during the development program).



TABLE 1-1

PROGRAM OBJECTIVES

TASK I: Parametric Study Over Power Range

- o Define optimum system characteristics
- o Determine system performance sensitivity to individual design parameters
- o Assess compatibility of system designs and existing technology base

TASK II: Off-Design Performance

- o Determine reference system design point
- o Select reference system components
- o Determine the number of unique engine designs required to cover power level spectrum

TASK III: Preliminary Design

- o Compressor design integration with alternator heat exchanger
- o Gas bearings (minimize losses)
- o Thermal design (gas cooling)
- o Alternator - EM design and motor starting capability



The objectives of the program were to be pursued while constraining the study within certain fundamental guidelines. These guidelines were specified by the NASA, by the contract, and by technical directive during the conduct of the study. These are summarized in Table 1-2.

TABLE 1-2

MINI-BRAYTON STUDY GUIDELINES

General

- o System heat derived from 1, 2 or 3 isotope heat source capsules, each having a 2400 watt capacity
- o Component sizing and performance predictions based upon empirical data base
- o System fixed weights supplied by the NASA
- o System parasitic losses (electronic) supplied by the NASA
- o Meteoroid criteria defined in NASA SP-8013
- o Armor protection--(Refer to Astronautics and Aeronautics, Power Systems for Space Flight Volume II, Academic Press, 1962, Pages 551-579).

Specific

- o Assume turbine inlet temperature = 1600°F
- o Assume use of a gas radiator with stainless steel tubes
- o Assume use of foil gas bearings
- o Alternator hot spot limit = 400°F
- o Radiator sink temperature = -10°F



1.3 Conclusion and Summary Data

1.3.1 Conclusions

The conclusions drawn from this study have been quite encouraging. All of the key design goals, summarized in Table 1-3, were achieved.

Table 1-4 shows the results of the study to determine development risk. Forced variation in individual parameters were made by using the off-design computer model of the system. In every instance, a performance loss was prevented by assuming that the radiator surface area would be increased (resulting in a lower compressor inlet temperature). Hence, radiator size and weight can be used as a common index for assessing system weight penalty. It can be seen that for all of the parameters, the resulting weight penalty is modest.

All Task III key design goals were achieved, as summarized in Table 1-5.

TABLE 1-3

TASK I AND TASK II CONCLUSIONS

- o A single set of hardware would operate efficiently over the entire power range
- o The best design point was at the three heat source capsule condition
- o Any mission requiring up to approximately 2500 watts could be satisfied with a single system comprised of the reference recuperator and Brayton Rotating Unit (BRU) hardware and the reference radiator or a space radiator sized specifically for the mission. (The reference components are described in Section 1.3.2).
- o The development risk involved in the selected design is low, both in terms of compatibility of design with existing technology and in program impact due to component performance variation.



TABLE 1-4
DEVELOPMENT RISK ASSESSMENT

	<u>Design Value</u>	<u>Potential Deviation</u>	<u>*Penalty (lbs)</u>
T_1	536	--	--
η_c	76.1	-2%	16
η_t	83.2%	-2%	32
β	0.980	-0.012	11
E_r	0.975	-0.010	12
r_c	1.50	--	--
N	52000 rpm	--	--
η_g	0.917	-0.010	2
N_{sc}	0.07	--	--
T_{SINK}	450	+10°R	5

*Weight increase due to enlarging radiation surface



TABLE 1-5

TASK III CONCLUSIONS

- HIGH LEVEL OF CONFIDENCE IN DESIGN
 - Gas cooling of alternator
 - No critical speed problems
 - Hydrodynamic gas bearings
- ONE ROTATING UNIT CAN COVER 0.5 - 2.0 KW_e APPLICATIONS
- HIGH RELIABILITY
- HIGH EFFICIENCY
- LOW COST DESIGN



1.3.2 Summary Data

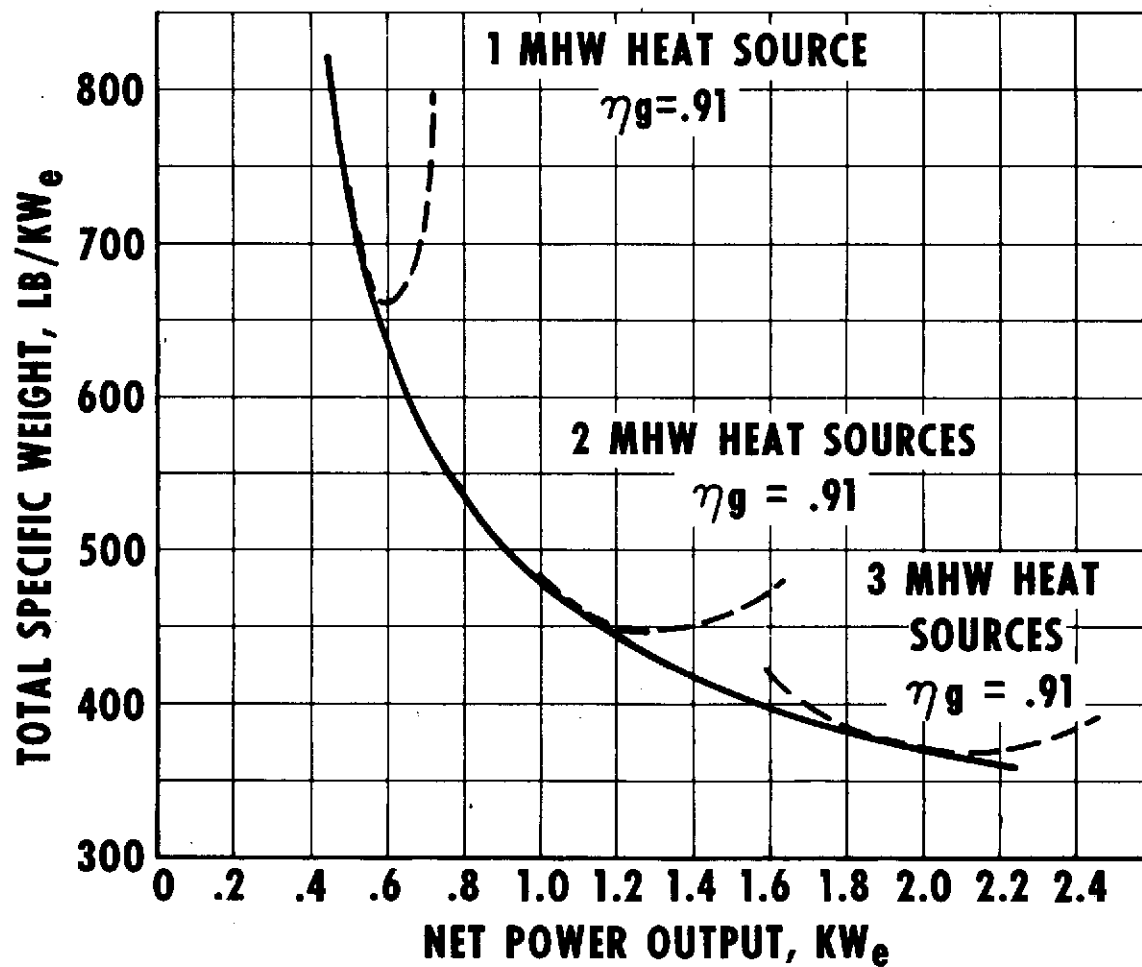
The initial part of the study was intended to not only result in selection of a specific reference cycle but to also define the general characteristics of low output closed Brayton cycle systems. Figure 1-4 shows the general case data. The line which is an envelope of and is tangent to the lines defining the minimum specified weight for a fixed number of heat source capsules itself defines the minimum specific weight achievable at any power level if the system is optimized at that power level. A real system, designed according to the constraints and guidelines imposed by this study, cannot be designed to fall below the general boundary shown.

Figure 1-5 shows the results of a study to define systems using a basic configuration (fixed hardware) of CRU and recuperator, combined with various numbers of multi-hundred watt heat sources and radiators of various sizes and configurations. In this study, the radiator parameters that were varied were surface area and radiator core pressure drop. This data is very useful for mission planning. For a given mission, one can determine the power conversion system weight and the number of heat source capsules required to produce the desired electrical power.

The reference cycle resulting from the study is also plotted in Figure 1-5. The complete definition of the reference cycle is shown in Figure 1-6. The off design performance and weight summaries of the reference design are presented in Table 1-6 and 1-7, respectively.



SYSTEM SENSITIVITY TO RECUPERATOR EFFECTIVENESS



MS 3063-8

FIGURE 1-4



RADIATOR CONFIGURATION OPTIMIZATION

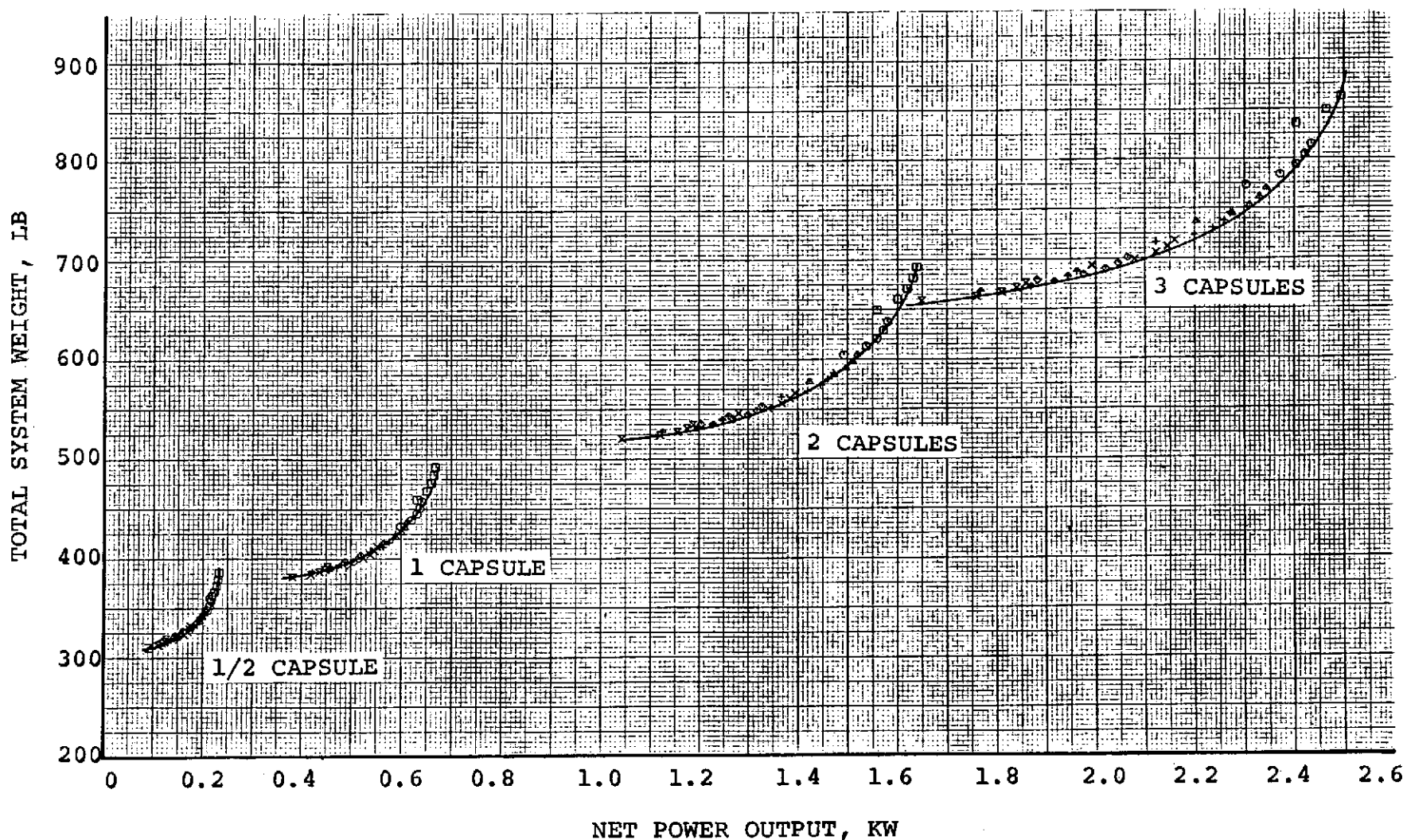


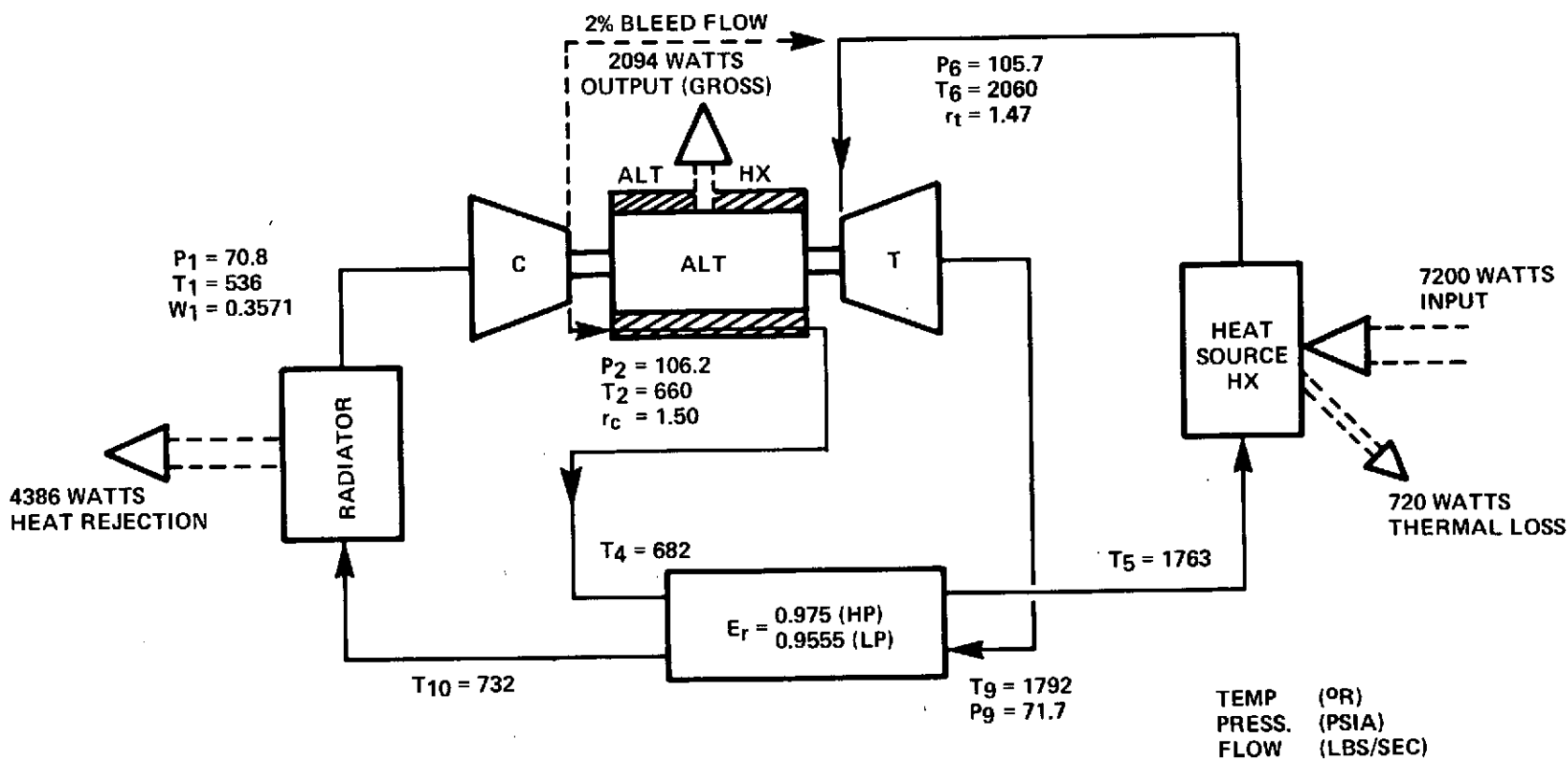
FIGURE 1-5



THREE CAPSULE MINIMUM SPECIFIC WEIGHT REFERENCE CYCLE

SHAFT SPEED 52,000 RPM
BEARING LOSS 241 WATTS
ALT. WINDAGE 68 WATTS
ALT. EFFICIENCY 91.7%

RADIATOR AREA 125 FT²
CONTROL POWER 81 WATTS
NET OUTPUT 2013 WATTS



MS 3063-9

FIGURE 1-6



The design and performance details for the reference system radiator, recuperator, and CRU are summarized in Tables 1-8, 1-9, and 1-10.

TABLE 1-6
SYSTEM PERFORMANCE AT THREE POWER LEVELS

	<u>Number of Capsules</u>		
	3	2	1
Alternator output power, kw	2.094	1.443	0.633
Control power, kw	0.082	0.069	0.053
Net cycle efficiency, percent	28.0	28.6	24.2
Compressor efficiency, percent	76.1	76.1	75.8
Turbine efficiency, percent	83.2	82.0	79.7
Alternator efficiency, percent	91.7	92.1	90.2
Lost pressure ratio parameter (β)	0.98	0.975	0.961
Recuperator effectiveness (E_r)	0.975	0.980	0.981
Compressor inlet temperature, °R	536	502	469
Compressor inlet pressure, psia	70.8	44.2	21.0
Compressor pressure ratio	1.5	1.54	1.59
Bearing loss, kw	0.241	0.209	0.180
Windage loss, kw	0.068	0.047	0.025
Compressor flow rate, lb/sec	0.357	0.236	0.117
Turbine inlet temperature, °R	2060	2060	2060
CRU speed, rpm	52,000	52,000	52,000



TABLE 1-7
SYSTEM WEIGHT FOR THREE POWER LEVELS

	<u>Number of Capsules (wt in lbs)</u>		
	<u>3</u>	<u>2</u>	<u>1</u>
Combined rotating unit	31	31	31
Recuperator	106	106	106
Radiator	160	160	160
Ducting (hot end only)	12	10	8
Capsule and HSHX	306	204	102
Super insulation	78	72	66
Structure	55	53	51
PLR	20	15	10
Electronics	<u>15</u>	<u>15</u>	<u>15</u>
TOTAL	783	666	549



Fixed weights to be used in system specific weight estimates are:

Number of MHW capsules	1	2	3
Capsule and HSHX weight (lb)	102	204	306
Super insulation weight (lb)	66	72	78
Structure weight (lb)	51	53	55
Parasitic load resistor weight (lb)	10	15	20
Electonrics weight (lb)	15	15	15

Cycle efficiency and system specific weight had been defined as:

Gross cycle efficiency =

$$\frac{\text{Gross power at alternator terminals}}{\text{Heat input to cycle working fluid}}$$

System specific weight (lb/kw) =

$$\frac{\text{Total of variable weights}}{\text{Gross power at alternator terminals}}$$

To achieve consistency in reporting efficiency and specific weight, the following terminology and computing methods were adopted:

Buss bar efficiency =

$$\frac{\text{Power at alternator terminals} - \text{electronic losses}}{2400 \text{ watts} \times \text{number of MHW capsules}}$$

Specific weight (lb/kw) =

$$\frac{\text{Total weight (including fixed weights)}}{\text{Power at alternator terminals} - \text{electronic losses}}$$



TABLE 1-8

MINI-BRU REFERENCE RADIATOR
STAINLESS STEEL/ALUMINUM ARMOR AND FINS

Total area, ft ²	125.3	
Gas flow rate, lb/sec	0.3573	(XeHe 83.8)
Tube length, ft	8.3205	
Radiator diameter, ft	4.8	
Inlet temperature, °F	733.1	
Outlet temperature, °F	536	
SS tube wall thickness, in.	0.030	
Fin thickness, in.	0.040	
Armor thickness, in.	0.0716	
Number of tubes	41	
Tube spacing, in.	4.4	
Header dimension, in.	1.5224	
Header weight, lb	33.1	
Core weight, lb*	117.6	
Total weight, lb	160.3	
Sink temperature, °F	450	

*Includes stainless steel tubes, fins, coating,
and armor.



TABLE 1-9

MINI-BRAYTON
REFERENCE RECUPERATOR
STAINLESS-STEEL/NICKEL
PLATE-FIN CONFIGURATION

Weight, lb	106
Width, in.	5.023
Height, in.	9.646
Core length, in.	24.427
Overall length, in.	28.777
Splitter plate thickness, in.	0.008
Side plate thickness, in.	0.100
Low pressure side	
Splitter plate spacing, in.	0.101
Fin thickness, in.	0.004
Fin length, in.	0.100
Number of fins per in.	20
High pressure side	
Splitter plate spacing, in.	0.101
Fin thickness, in.	0.004
Fin length, in.	0.100
Number of fins per in.	20
Total fractional pressure drop	0.006968



TABLE 1-10
REFERENCE CRU DATA

Total CRU weight, lb	31
CRU rotating speed, rpm	52,000
<u>Turbine Data</u>	
Diameter, in.	3.0
Pressure ratio	1.47
Efficiency	0.832
<u>Compressor Data</u>	
Diameter, in.	2.3
Pressure ratio	1.50
Efficiency	0.761
Foil journal bearing diameter, in.	1.0
Thrust bearing ID, in.	1.0
Thrust bearing OD, in.	1.75
<u>Alternator Physical Data</u>	
Total weight, lb	10.42
Rotor weight, lb	1.053
Rotor diameter, in.	1.5
Rotor length, in.	3.72
Frame OD, in.	4.14
Frame width, in.	3.38
Stack length, in.	0.72
<u>Alternator EM Efficiency</u>	0.917



2. TASK I - PARAMETRIC STUDY AND TASK II - OFF DESIGN STUDY

2.1 Summary of Analytical Approach

Details of the Task I and Task II analyses, including the method used to achieve system optimization and a description of the analytical tools employed are described in this section. The rationale leading to the selection of the reference system and many design approaches investigated prior to the final selection are also discussed. The component selection justification is documented, as well as a review of the important design considerations for the various components.

The data generated during this part of the study is the result of the application of a cascading series of the most sophisticated computer design tools existing in the aerospace industry. The automated procedure goes from basic thermodynamic cycle optimization to detail component sizing to automatic machine plotting of the results.

2.1.1 System Specifications

The specifications designated by the NASA defined the parametric and geometric ranges to be covered in the study. These were:

(a) Compressor

- (1) Cycle temperature ratio: T_1/T_2
- (2) Pressure ratio: 1.6 through 2.2
- (3) Slip factor (defined as $\frac{\text{tangential gas velocity}}{\text{wheel tip speed}}$)



(b) Turbine

(1) Specific speed

(2) Lost pressure ratio: β factor, where $\beta = r_t/r_c$
 $r_t/r_c = 0.92, 0.94, 0.96$

(c) Working Fluid

(1) He/Xe, Mw = 60, 83.8, 39.94

(2) Argon

(3) Krypton

(d) Rotating speed = 48,000 rpm - 60,000 rpm

(e) Recuperator

(1) Various core geometries

(2) Effectiveness = 0.90, 0.925, 0.95

(f) Radiator

(1) With and without internal fins in tubes

(2) Various internal fin geometries

(3) All aluminum construction

(4) Stainless steel tubes and headers with aluminum armor
and fins



(g) Heat source H/X (*): Vary cores

(h) Alternator H/X: Vary cores

2.1.2 Data Requirements

The study results were to include but not be limited to the following system and component characteristics:

(a) Turbine and Compressor

(1) Size and weight

(2) Efficiency

(3) Pressure ratio

(4) Specific speed

(b) Alternator

(1) Physical dimensions

(2) Electromagnetic performance

(c) System

(1) Overall efficiency

(2) Component weights and sizes

*Sketches and design considerations for the heat source were furnished by the NASA.



- (3) Molecular weights (fluid)
- (4) Duct sizes
- (5) Flow rates
- (6) State points (pressure and temperature)
- (d) Windage Losses
- (e) Bearing Losses

2.1.3 Analytical Procedure or Methodology

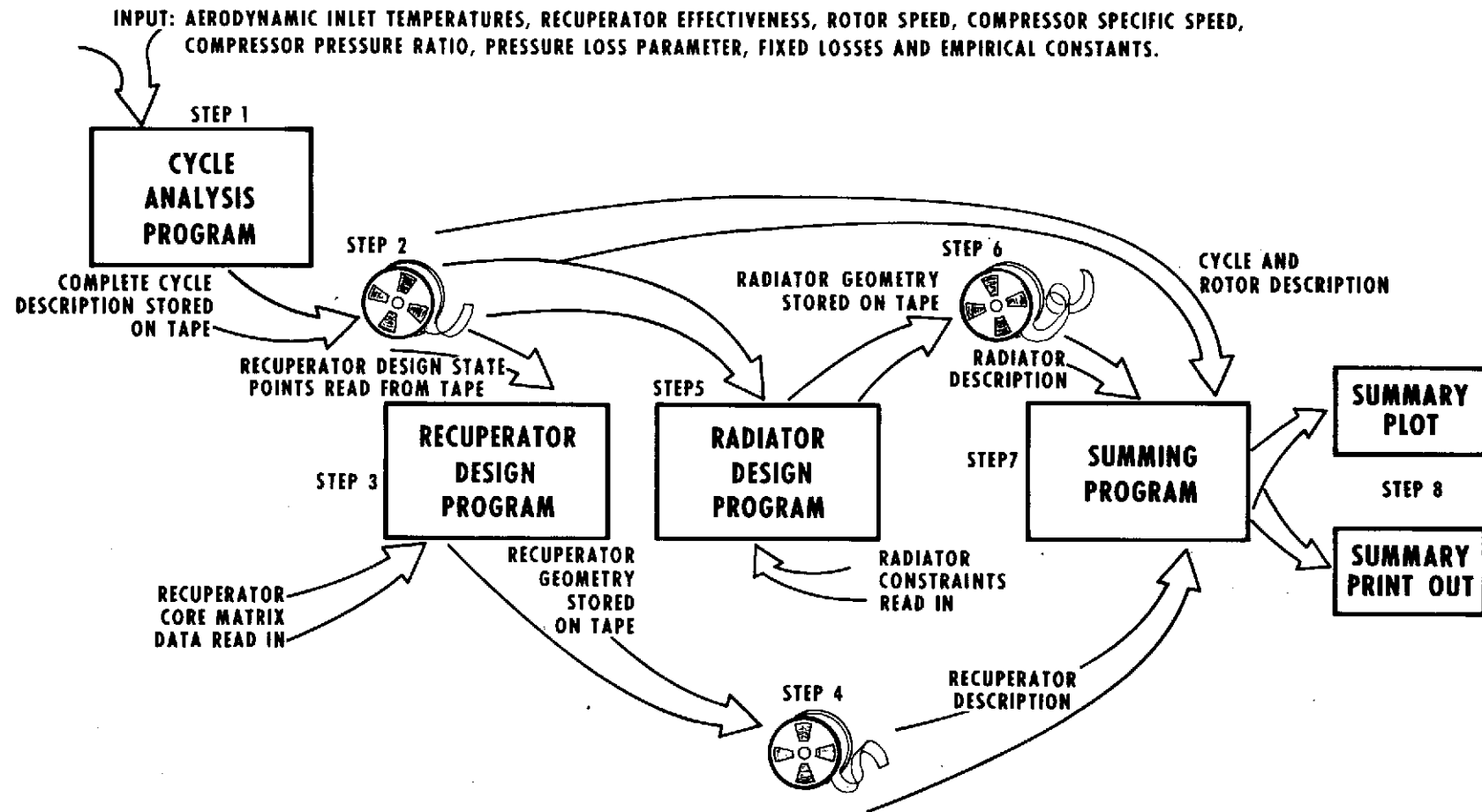
The study progresses from wide range parametric studies in the initial phases, resulting in later elimination by inspection of a large number of systems.

After the envelope was defined for minimum weight systems over the entire power level range (refer to Figure 1-4), only specific system designs lying close to that envelope were investigated further.

Several of AiResearch's component and cycle design computer programs were merged in order to define the geometry and performance of a complete closed, recuperated Brayton cycle power system. During the Task I study, literally tens-of-thousands of complete systems were designed, sorted and machine-plotted in order to define the minimum envelope of weight and power. Figure 2-1 describes, schematically, the analytical procedure:



POWER SYSTEM DESIGN METHODOLOGY



MS 3063-10

FIGURE 2-1



Step 1

As can be seen in Figure 2-1, the input to the cycle design portion of the analysis consists mainly of the aerodynamic, thermodynamic and empirical specifications of the desired systems. These values include:

- o Desired recuperator effectivenesses (up to 12 values) - E_r
- o Desired rotor speed (up to 12 values) - N
- o Desired compressor specific speed (up to 12 values) - N_{s_c}
- o Desired compressor pressure ratio (up to 12 values) - r_c
- o Desired compressor inlet temperature (up to 12 values) - T_1
- o Desired turbine inlet temperature - T_6
- o Desired output power or input power (up to 12 values) - PWR
- o Compressor bleed fraction (up to 12 values) - BL
- o Desired pressure loss distribution (up to 12 values) - $\Delta P/P$
- o Predicted alternator efficiency - η_g
- o Specification of working fluid properties
- o Desired compressor slip factor, turbine velocity factor and turbine diffuser recovery - SC, F_v, CPR
- o Bearing power loss - PWR_b
- o Alternator sizing coefficient - C_g



Inherent to the cycle program are also empirical data bases for turbomachinery efficiency predictions (based on Reynold's number, size, pressure ratio and specific speed) and for alternator windage calculation.

When a cycle definition is complete, the following values have been generated:

- o Compressor, turbine and alternator diameter
- o Compressor, turbine and alternator efficiency
- o Rotating group weight
- o Gross alternator electrical output
- o Alternator windage
- o Gross cycle efficiency
- o All state point temperatures, pressures and fractional pressure drops

Step 2

As the data compilation for each computed cycle is completed, the pertinent parameters are written out to magnetic tape storage for further processing upon completion of the cycle analysis. When the desired range of cycle parameters has been explored and the output from each stored, the analysis proceeds into the recuperator design program. The output for each cycle can also be printed at this point. A typical print-out is shown in Figure 2-2.



TYPICAL CYCLE DESIGN PROGRAM OUTPUT

CASE NUMBER 3

GENERATOR OUTPUT (KW) .20772E+01	GEN. ROTOR DIAMETER (IN.) .15000E+01	GROSS SHAFT POW. OUT. (KW) .25849E+01	GEN. WINDAGE LOSS (KW) .72613E-01	CYCLE EFFICIENCY (----) .32068E+00
RECUPERATOR EFFECT. (----) .97500E+00	CRUI. ROTATION. SPEED (RPM) .52000E+05	BLEED FLOW FRACTION (----) .20000E-01	COMP. TOT. TEMP. RATIO (----) .12313E+01	PRESS. LOSS FACTOR (PRT/PRC) .98000E+00
COMP. ADIAB. EFF. (----) .76127E+00	COMP. ROTOR DIAM. (IN.) .23003E+01	COMP. PRESS. RATIO (----) .15000E+01	REQD. SURFACE FINISH (MICRO IN.) .10226E+02	COMP. SPEC. SPEED (----) .70000E-01
TURB. ADIAB. EFF. (----) .83161E+00	TURB. ROTOR DIAM. (IN.) .28543E+01	TURB. PRESS. RATIO (----) .14700E+01	TURB. ROTOR TIP SPEED (FPS) .64762E+03	TURB. SPEC. SPEED (----) .54408E-01
COMP. + TURB. WEIGHT (LB.) .14619E+00	GEN. PER. REV. NO. (----) .30262E+05	COMP. MEAN SPEC. SP. (----) .44337E+02	COMP. EXPAN. EFF. (----) .87925E+00	GEN. DISK REV. NO. (----) .11351E+07

COMPONENT	FLOW (LB/SEC)	INLET TEMP. (R)	INLET PRESS. (PSIA)	OUTLET TEMP. (R)	OUTLET PRESS. (PSIA)
COMPRESSOR	.35730E+00	.53600E+03	.70826E+02	.65998E+03	.10624E+03
RECUPERATOR	.35016E+00	.68319E+03	.10589E+03	.17638E+04	.10553E+03
HEATER	.35016E+00	.17638E+04	.10553E+03	.20600E+04	.10535E+03
TURBINE	.35016E+00	.20600E+04	.10535E+03	.18152E+04	.71669E+02
RECUPERATOR	.35730E+00	.17921E+04	.71669E+02	.73311E+03	.71301E+02
COOLER	.35730E+00	.73311E+03	.71301E+02	.53600E+03	.71063E+02

FIGURE 2-2



Step 3

The core matrix data (i.e., heat transfer and pressure drop maps) and the desired margin for heat transfer and pressure drop are read-in. From the data stored for each computed cycle, the variables pertinent to recuperator design effectiveness (inlet conditions and gas properties) are obtained. One or more recuperators are designed for each computed cycle.

Step 4

The recuperator design parameters for each design are, in turn, stored on another tape. The individual recuperator designs can be called for and printed at this point. A typical print-out is shown in Figures 2-3 and 2-3 Cont'd.

Step 5

The design of the gas flow radiators proceeds in a manner similar to that for the recuperator design.

Step 6

The radiator design program output is likewise stored on a third tape, and/or printed out in the format presented in Figure 2-4.

Step 7

The summation program serves the function of merging the individual component designs and cycles into a total system representation. The summation program also sizes the minor components, such as ducting, and controls, for each system.



TYPICAL RECUPERATOR DESIGN PROGRAM OUTPUT

INPUT GEOMETRY FOR COUNTER FLOW HEAT EXCHANGER DESIGN PROGRAM

MINI-BRU RECUPERATOR

	GAS	UA MARGIN (-----)	DP MARGIN (-----)	F MARGIN (-----)	J MARGIN (-----)		
L.P. SIDE -----	XE/HE83	.2000	.2000	-0.0000	-0.0000		
H.P. SIDE -----	XE/HE83	.2000	.2000	-0.0000	-0.0000		
	NUMBER (PER INCH)	THICKNESS (INCHES)	SPACING (INCHES)	EQ. LENGTH (INCHES)	DENSITY (LB/CU-IN)	CONDUCTIVITY (BTU/HR-FT-R)	
L.P. SIDE FINS -----	20.00	.004	.101	.050	.297	6.000	
H.P. SIDE FINS -----	20.00	.004	.101	.050	.297	6.000	
L.P. SIDE HEADER FINS -----	10.00	.006			.297		
H.P. SIDE HEADER FINS -----	10.00	.006			.297		
SPLITTER PLATES -----		.008			.297	10.000	
WRAP-UP MATERIAL -----		.060			.297	10.000	
SPACER BARS -----		.100			.297	10.000	
BRAZE MATERIAL -----		.001			.297	10.000	
	FLOW AREA RATIO (-----)	SURFACE AREA/VOLUME (1/IN.)	FIN AREA/VOLUME (1/IN.)	PLATE AREA/VOLUME (1/IN.)	HYDRAULIC DIAMETER (INCHES)		
L.P. SIDE FINS -----	.40936	26.239	17.798	8.440	.06241		
H.P. SIDE FINS -----	.40936	26.239	17.798	8.440	.06241		
L.P. SIDE HEADER FINS -----	.40963	17.339	8.716	8.624	.09450		
H.P. SIDE HEADER FINS -----	.40963	17.339	8.716	8.624	.09450		
	ENTRY LOSS (-----)	TURNING LOSS (-----)	EXIT LOSS (-----)	TURNING ANGLE (DEGREES)			
L.P. SIDE HEADER FINS -----	.40400	.18000	1.00000	30.00			
H.P. SIDE HEADER FINS -----	.40400	.60000	1.00000	60.00			

FIGURE 2-3

TYPICAL RECUPERATOR DESIGN PROGRAM OUTPUT



PLATE FIN CONVECTION DESIGN
41N1-440 RECUPERATOR

LOW PRESSURE SIDE						HIGH PRESSURE SIDE					
EFFECTIVENESS	INLET TEMPERATURE	GAS IS XE/HEAT	INLET TEMPERATURE	GAS IS XE/HEAT	INLET TEMPERATURE	EFFECTIVENESS	INLET TEMPERATURE	GAS IS XE/HEAT	INLET TEMPERATURE	GAS IS XE/HEAT	INLET TEMPERATURE
0.9552E+00	17321E+04	(DEG. R)	173236E+03	(DEG. R)	17321E+04	0.9750E+00	173236E+03	(DEG. R)	17321E+04	(DEG. R)	17321E+04
FLUX RATE	INLET PRESSURE	FLUX RATE	INLET PRESSURE	FLUX RATE	INLET PRESSURE	FLUX RATE	INLET PRESSURE	FLUX RATE	INLET PRESSURE	FLUX RATE	INLET PRESSURE
0.75730E+00	0.7170E+02	(LB/SEC)	0.47390E-02	(DP/P)	0.7170E+02	0.75016E+00	0.47390E-02	(LB/SEC)	0.7170E+02	(DP/P)	0.7170E+02
FIN EFFECTIVENESS	REYNOLDS NO.	(-----)	0.26537E+00	(-----)	0.26537E+00	0.42925E+03	0.42925E+03	(-----)	0.26537E+00	(-----)	0.26537E+00
0.86645E+00	0.2745E+03	(-----)	0.26537E+00	(-----)	0.26537E+00	0.42925E+03	0.42925E+03	(-----)	0.26537E+00	(-----)	0.26537E+00
M.T. CONDUCTANCE	VISCOSITY	(BTU/SEC. DEG. R)	0.32473E-04	(LB/FT-SEC)	0.32473E-04	0.32086E-04	0.32086E-04	(LB/FT-SEC)	0.32473E-04	(LB/FT-SEC)	0.32473E-04
0.16418E+01	0.32473E-04	(BTU/SEC. DEG. R)	0.5920E-01	(BTU/LB-DEG. F)	0.5920E-01	0.16100E+01	0.16100E+01	(BTU/LB-DEG. F)	0.32473E-04	(BTU/LB-DEG. F)	0.32473E-04
HEADQU WEIGHT	FIN WEIGHT	(LB)	0.13520E+00	(LB)	0.13520E+00	0.15445E+01	0.15445E+01	(LB)	0.13520E+00	(LB)	0.13520E+00
0.15445E+01	0.13520E+00	(LB)	0.13520E+00	(LB)	0.13520E+00	0.15445E+01	0.15445E+01	(LB)	0.13520E+00	(LB)	0.13520E+00
TOTAL WEIGHT	PLATE WEIGHT	(LB)	0.14710E+02	(LB)	0.14710E+02	0.16272E+02	0.16272E+02	(LB)	0.14710E+02	(LB)	0.14710E+02
0.14501E+03	0.24093E+02	(LB)	0.14710E+02	(LB)	0.14710E+02	0.16272E+02	0.16272E+02	(LB)	0.14710E+02	(LB)	0.14710E+02
CORE LENGTH	CORE WIDTH	(INCHES)	0.96455E+01	(INCHES)	0.96455E+01	0.28177E+02	0.28177E+02	(INCHES)	0.96455E+01	(INCHES)	0.96455E+01
0.24427E+02	0.50227E+01	(INCHES)	0.96455E+01	(INCHES)	0.96455E+01	0.28177E+02	0.28177E+02	(INCHES)	0.96455E+01	(INCHES)	0.96455E+01
TOTAL PRESSURE DROP						TOTAL PRESSURE DROP					
0.69680E-02	0.69680E-02	(DP/P)	0.69680E-02	(DP/P)	0.69680E-02	0.69680E-02	0.69680E-02	(DP/P)	0.69680E-02	(DP/P)	0.69680E-02
0.69680E-02	0.69680E-02	(DP/P)	0.69680E-02	(DP/P)	0.69680E-02	0.69680E-02	0.69680E-02	(DP/P)	0.69680E-02	(DP/P)	0.69680E-02

FIGURE 2-3 CONT'D.



TYPICAL RADIATOR DESIGN PROGRAM OUTPUT

CYCLE NUMBER 3

MASS FLOW (LB/SEC) .35730E+00	CORE PRESSURE DROP --- .36000E-02	HEADER PRES. DROP --- .14660E-02	RET. DUCT PRES. DROP --- .27800E-03	SUP. DUCT PRES. DROP --- .27800E-03	INLET TEMP. (R) .73311E+03
OUTLET TEMP. (R) .53600E+03	NO. INTERNAL FINS ---- -0.	SINK TEMP. (R) .45000E+03	TUBE LENGTH (FT) .83205E+01	NO. HEADER PAIRS ---- .10000E+01	EXTERNAL FIN EFF. ---- .96455E+00
TUBE INT. DIAM. (IN) .31667E+00	TOTAL WEIGHT (LB) .16029E+03	CORE WEIGHT (LB) .11764E+03	HEADER WEIGHT (LB) .33086E+02	DUCT WEIGHT (LB) .95645E+01	NUMBER OF TUBES ---- .41044E+02
RADIATOR AREA (SQ FT) .12526E+03	TUBE SPACING (IN) .44014E+01	EXT. FIN THICK. (IN) .40000E-01	INLET WALL TEMP. (R) .72142E+03	OUTLET WALL TEMP. (R) .53395E+03	ARMOR THICKNESS (IN) .71583E-01
HEADER FLOW DIA. (IN) .15224E+01	FIN BASE TEMP. (R) .64777E+03	FILM COEFFICIENT (BTU/SEC SQ FT F) .65653E-02	REYNOLDS NO. ---- .21936E+05	RET. DUCT DIAM. (IN) .16170E+01	SUP. DUCT DIAM. (IN) .17477E+01
FLUID SPEC. HEAT (BTU/LB F) .59200E-01	VULNERABLE AREA (SQ FT) .13075E+02	INTERNAL FIN EFF. ---- .94856E+00	COATING WEIGHT (LB) .51764E+01	TUBE WEIGHT (LB) .49319E+02	FIN WEIGHT (LB) .64612E+02
.60000E-02					

APS-5440-R
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FIGURE 2-4



Step 8

The output of the summation program is presented in two forms. The first (refer to Figure 2-5) is a tabulation of one line summaries of cycle specifications and component geometry (including predicted system weight and radiator area). The second form of output is available in plot form. The initial form of the plots are the computed specified weight (including the variable weights, in lb/kw_e , of the BRU, recuperator, and radiator) for the system and the computed specific radiator area plotted versus gross cycle efficiency. This plotted output forms the basis for the determination of the boundary of minimum attainable system weight as a function of system performance and output. Figure 1-4 shows a typical boundary generated in this fashion. The basic philosophy employed during the study was that one can determine the point on such a boundary which best expresses the desired trade-off of cycle efficiency (fuel cost) and system weight (launch cost). Having selected a point on the boundary, one seeks to precisely define a system in this area such that no system variable can be perturbed in any manner without a resulting upward departure from the boundary. This philosophy implies that a variation of a single system parameter (referred to as a single dimension parametric study) should produce a trace (refer to Figure 2-6) of specific weight versus efficiency which is tangent to the previously defined lower boundary.

The philosophy of system selection was later expanded to include the predicted performance of a particular (fixed) system across the entire range of power output (off-design performance). This approach will be discussed in more detail later in this section.



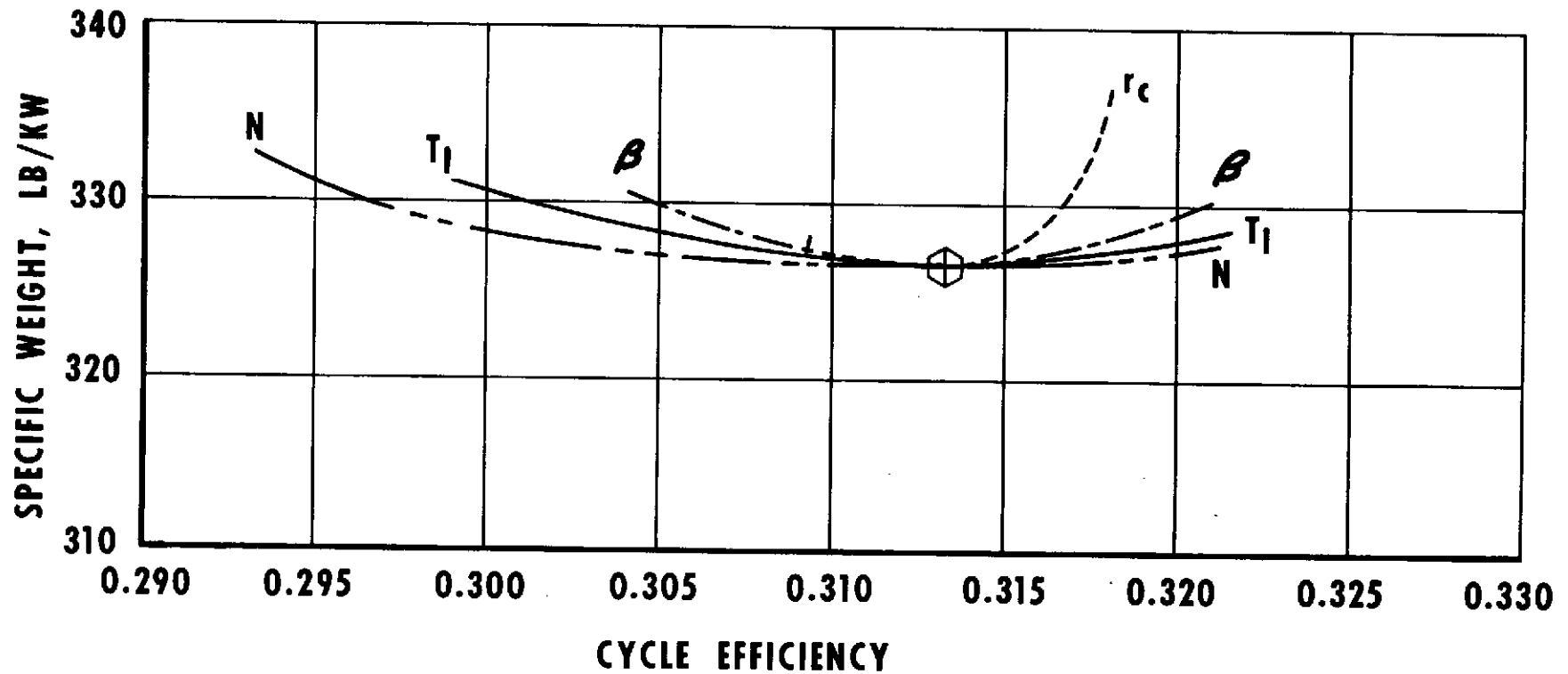
TYPICAL SYSTEM SUMMARY OUTPUT

CASE	MW	PRC	RPM	ER	PETA	NSC	WCRU	VCRU	WR	VR	WRAD	ARAD	WD	VD	WT	AT	ECY
1	84.	1.50	52000.	.96	.98	.07	23.3	.15	34.2	.34	160.3	124.59	11.3	.27	368.3	65.3	.2652
2	84.	1.50	52000.	.97	.98	.07	23.3	.15	39.8	.40	160.3	124.93	11.3	.27	363.1	64.0	.2711
3	84.	1.50	52000.	.98	.98	.07	23.3	.15	47.4	.48	160.3	125.26	11.3	.27	359.9	62.8	.2773

FIGURE 2-5



SINGLE DIMENSION PARAMETRIC STUDY RESULTS



MS 3063-5

FIGURE 2-6



2.1.4 Reference Design Selection

In order to select the recommended reference cycle, data was generated using both system design optimization computer programs and system off-design performance computer programs. Over 8,000 completely defined systems were designed to determine general system characteristics over the power range of interest. In addition to that part of the study, several other design approaches were examined to determine which "fixed hardware" system (i.e., a particular system design) had the most favorable performance characteristics over the entire power range and how the off-design performance of a particular system compared to the systems optimized at each power level.

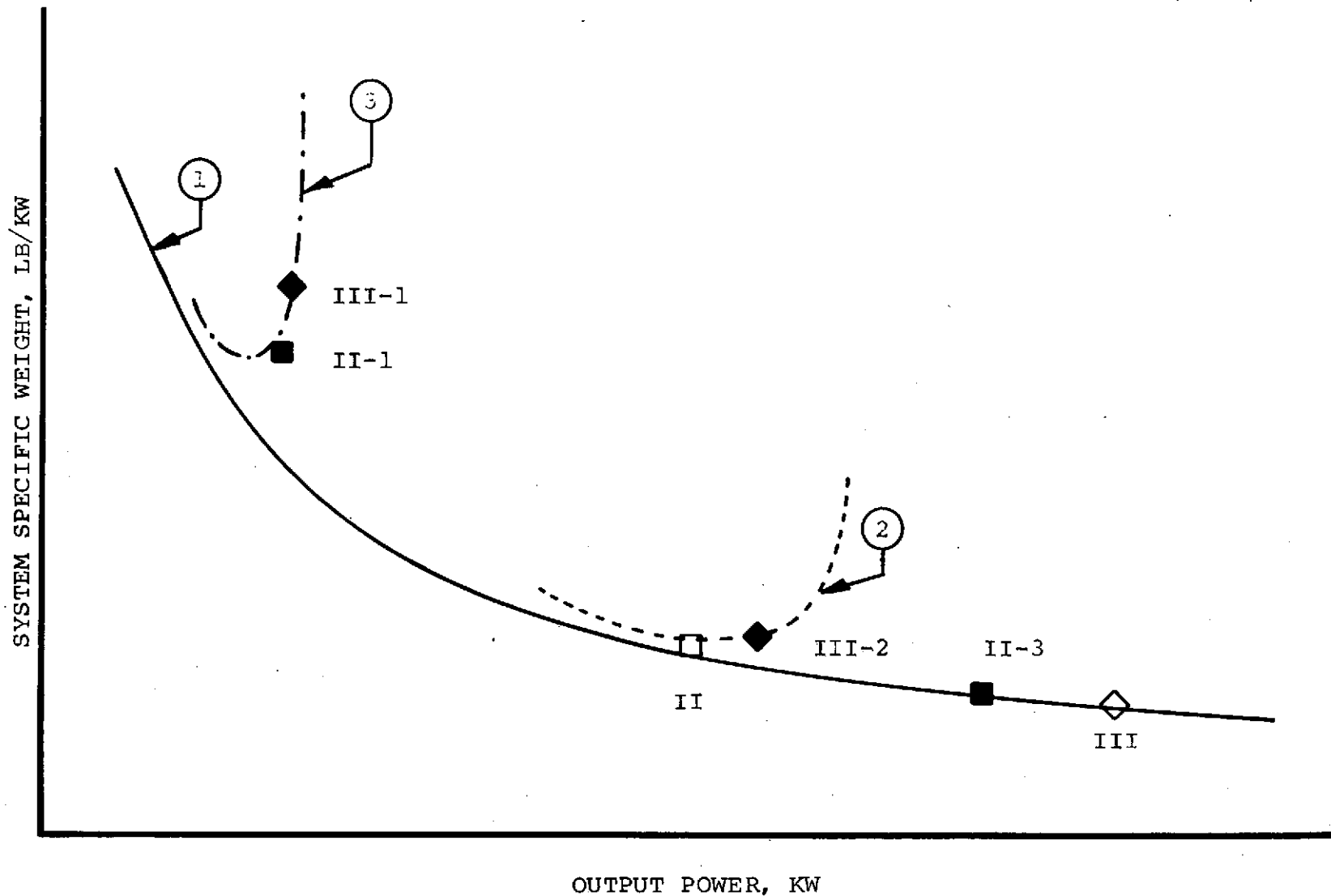
The trends of this investigation are summarized in Figure 2-7. The actual working curves were plotted on large (20 in. x 30 in.) graph paper and submitted to the study program monitor. To avoid confusion and to facilitate an explanation of the technical approach followed, the trends only are shown in Figure 2-7.

Line (1) defines the loci of minimum specific weight system designs as a function of power level. The particular designs indicated as II and III are those which are optimized (with regard to minimum specific weight) when two or three heat source capsules are used. Points III-2 and III-1 indicate the off-design performance of system III operating with two and one heat source capsules. Similarly, Points II-1 and II-3 denote the off-design performance of System II.

A similar exercise was not performed for the optimum system using one heat source because, due to the limited capability of the 0.5 kw_e alternator, it could not be made to produce 2 kw_e during off-design operation.



SYSTEM SPECIFIC WEIGHT CHARACTERISTICS



OUTPUT POWER, KW

FIGURE 2-7



Another possibility studied was a system assumed to be comprised of the reference system (III) rotating unit and recuperator but with varying size radiators. Lines (2) and (3) show the trends and an inspection of the data yields ample justification for the important conclusion that one CRU and recuperator, which would be designed specifically for the 3 heat source capsule case, would be quite suitable for use over the entire power range. This is a tremendous advantage since it precludes the need of multiple development efforts on several configurations of the same basic set of hardware.

The clinching argument for selection of the system optimized for the 3 heat source capsule case as the reference system was the results of a comparison study of the reference system with three other candidate specific system designs. The system designs compared included:

- (a) A system optimized for 0.5 kw_e . Since this system included a CRU with an alternator of limited output, multiple units would have to be employed in parallel to achieve 2.0 kw_e .
- (b) Approach (a) modified to include an oversized alternator. However the heat exchangers, sized for the low power condition, represent a limitation.
- (c) A system optimized to operate with 2 heat source capsules.

The system comparison results are shown in Table 2-1. The numbers represent an attempt to give some quantification to the comparison.



TABLE 2-1
SYSTEM COMPARISON MATRIX

	Development Risk	Efficiency-over Power Range	Weight	Cost	Power Capability	Reliability	Total
A. Optimized for 1 heat source capsule (multiple systems used for higher power)	1 ⚠	3	1 ⚠	1	1	3	10
B. Optimized for 1 heat source capsule with 2.0 kw _e alternator	1 ⚠	1 ⚠	3	2	2 ⚠	2	11
C. Optimized for 2 heat source capsules	2 ⚠	2	2	3	2	2	13
D. Optimized for 3 heat source capsules	3	3	2 ⚠	3	3	2	16

Merit Scale

- 3 - Most advantageous
- 2 - In between
- 1 - Least advantageous

- ⚠ The optimum 0.5 kw_e system has a smaller diameter compressor than the reference system (2.1 in. versus 2.3 in.)
- ⚠ Multiple units required to achieve 2.0 kw_e
- ⚠ Heat exchangers are undersized for higher power level applications, resulting in inefficiency
- ⚠ Larger system, somewhat oversized for low power application



3. TASK III - PRELIMINARY DESIGN OF ROTATING UNIT

3.1 Mechanical Design

The mini-BRU rotating group configuration consists of a single-stage compressor, turbine, and an alternator on a common shaft. The shaft is supported on gas-lubricated foil journal and thrust bearings with self-acting start-stop capability. The journal bearings are located in the alternator secondary air gap to minimize shaft length. Further, in order to minimize losses, the machine design specifically excludes seals; rather the journal and thrust bearings act as seals to prevent gas flow from the compressor to turbine ends.

3.1.1 Bearings

The general design requirements for the mini-BRU foil gas bearings are presented in Table 3-1. Based on these requirements, the bearing design was initiated by conducting a parametric analysis to determine the critical speed characteristic of the rotating group as a function of the foil bearing stiffness. Results of this analysis, presented in Figure 3-1, show that at low bearing spring rates both the first and second critical speed regimes are far below the 100 percent operating speed of 52,000 rpm. Also, the 160,000 rpm third critical is shown to be independent of spring rate and is far above even the runaway speed (limiting speed due to aerodynamic component performance characteristics) of 92,000 rpm.

Following the parametric analysis, a bearing spring rate of 3,300 lb/in. was selected, through successive iterations, as that yielding the most desirable shaft dynamics. The bearing load versus shaft speed plots are presented in Figure 3-2, for a shaft eccentricity of 0.0005 in. and journal diameter of 1.0 in. As may be noted from the figure, the first and second criticals (7,200 rpm and 9,200 rpm respectively) are significantly removed from the engine operating speed such that no critical speed problem is anticipated. The third critical lying far



FOIL BEARING GENERAL DESIGN REQUIREMENTS

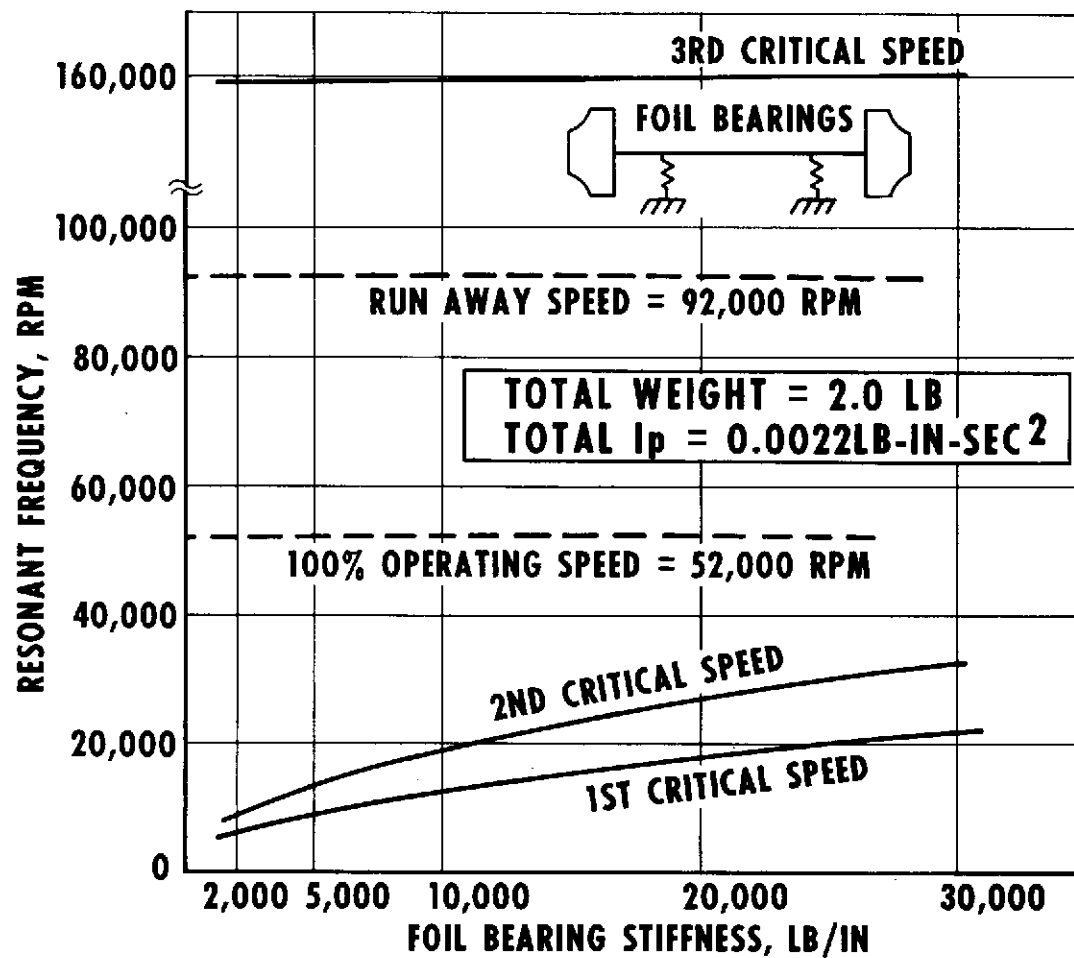
- **DESIGN SPEED = 52,000 RPM**
- **OPERATING TEMPERATURE = 450°F**
- **WORKING FLUID – (XE-HE) M.W. = 83.8**
- **AMBIENT PRESSURE**
 - = 24 PSIA**
 - = 95 PSIA**
- **ROTOR WEIGHT = 1.68 LB.**
- **LOW BEARING LOSS**
- **LOW DRAG TORQUE**

MS 3011-20

TABLE 3-1



CRITICAL SPEED VS BEARING SPRING RATE

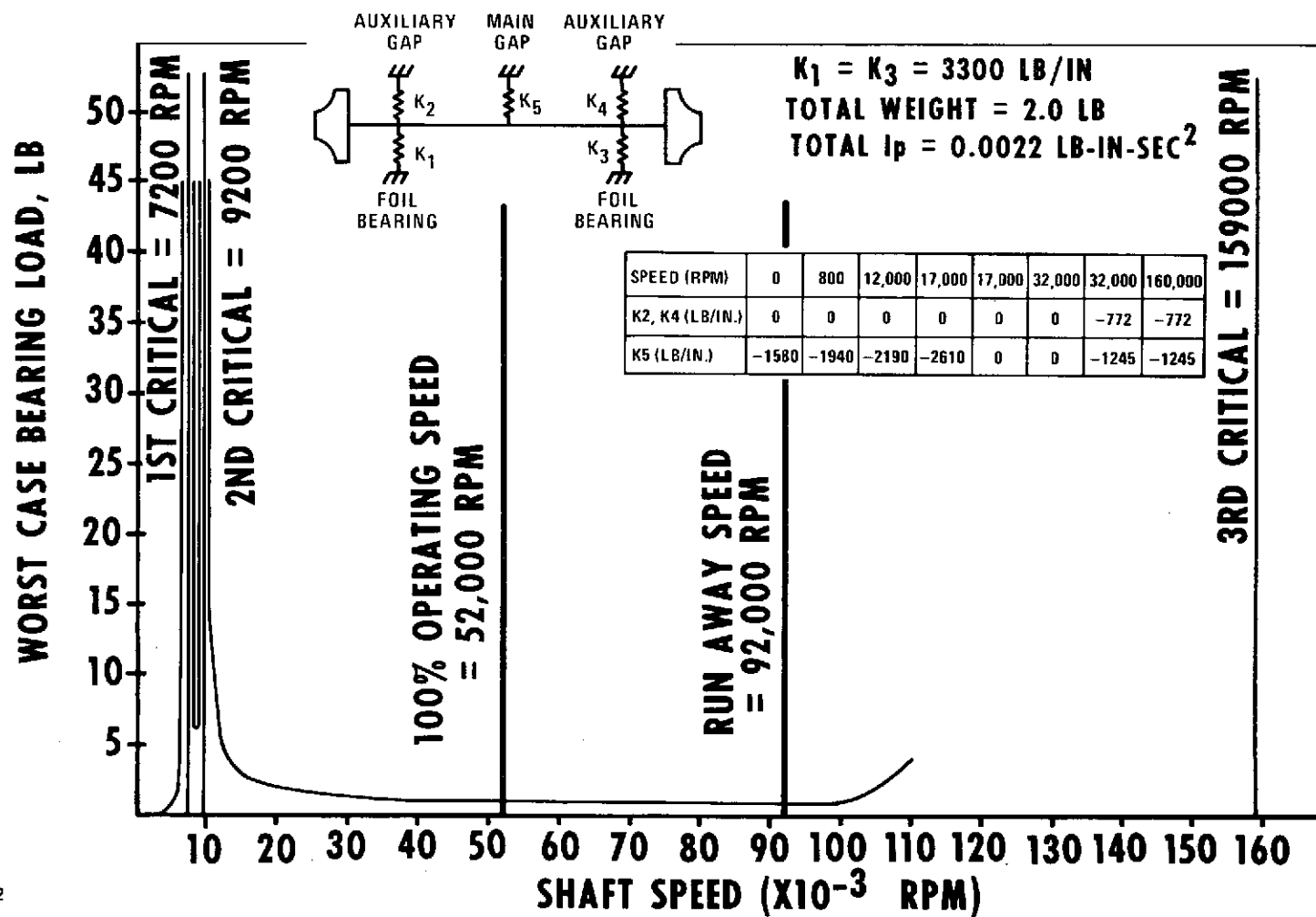


MS 3063-11

FIGURE 3-1



BEARING LOADS



MS 3063-12

FIGURE 3-2



above the runaway speed provides the potential for simplification of the overspeed control design. The table at the top of Figure 3-2 summarizes the unbalanced magnetic forces, expressed as "negative" bearing spring rates, for the alternator in the motor start mode (0 through 17,000 rpm), during bootstrap operation (17,000 through 32,000 rpm) and in the normal alternator mode producing electrical power (32,000 through 160,000 rpm). Data for the table was generated in the alternator magnetic unbalance analyses for motor start and alternator operation (refer to Figures 3-21 and Table 3-14, respectively).

The resulting foil journal bearing design best satisfying the load requirements and yielding minimum losses is summarized in Table 3-2.

In order to properly size the foil thrust bearing for minimum power loss, a turbine and compressor thrust balance analysis was conducted. Figure 3-3 presents the estimated turbine thrust as a function of back face slip factor and percent turbine scallop. As may be noted from the figure a wide variance in thrust may be obtained by changing the turbine shroud clearance (and hence slip factor) and/or decreasing the turbine back face area through material removal. Further, Figure 3-4 shows the effect of back face slip factor and the addition of compressor lip on net compressor thrust. Therefore, by selecting the proper combination of shroud clearances, turbine scallop and compressor lip height, an optimum system thrust balance may be achieved leading to a minimum loss thrust bearing design. This design is summarized in Table 3-3.

The journal and thrust bearing power losses were estimated for bearing cavity pressures of 24 and 95 psia (corresponding to the 1 and 3 capsule power level conditions). These results are presented in Table 3-4. Estimated performance for the journal and thrust bearings, plotted as power loss (watts) versus unit load (psi), is shown in figures following Table 3-4.



JOURNAL BEARING DESIGN

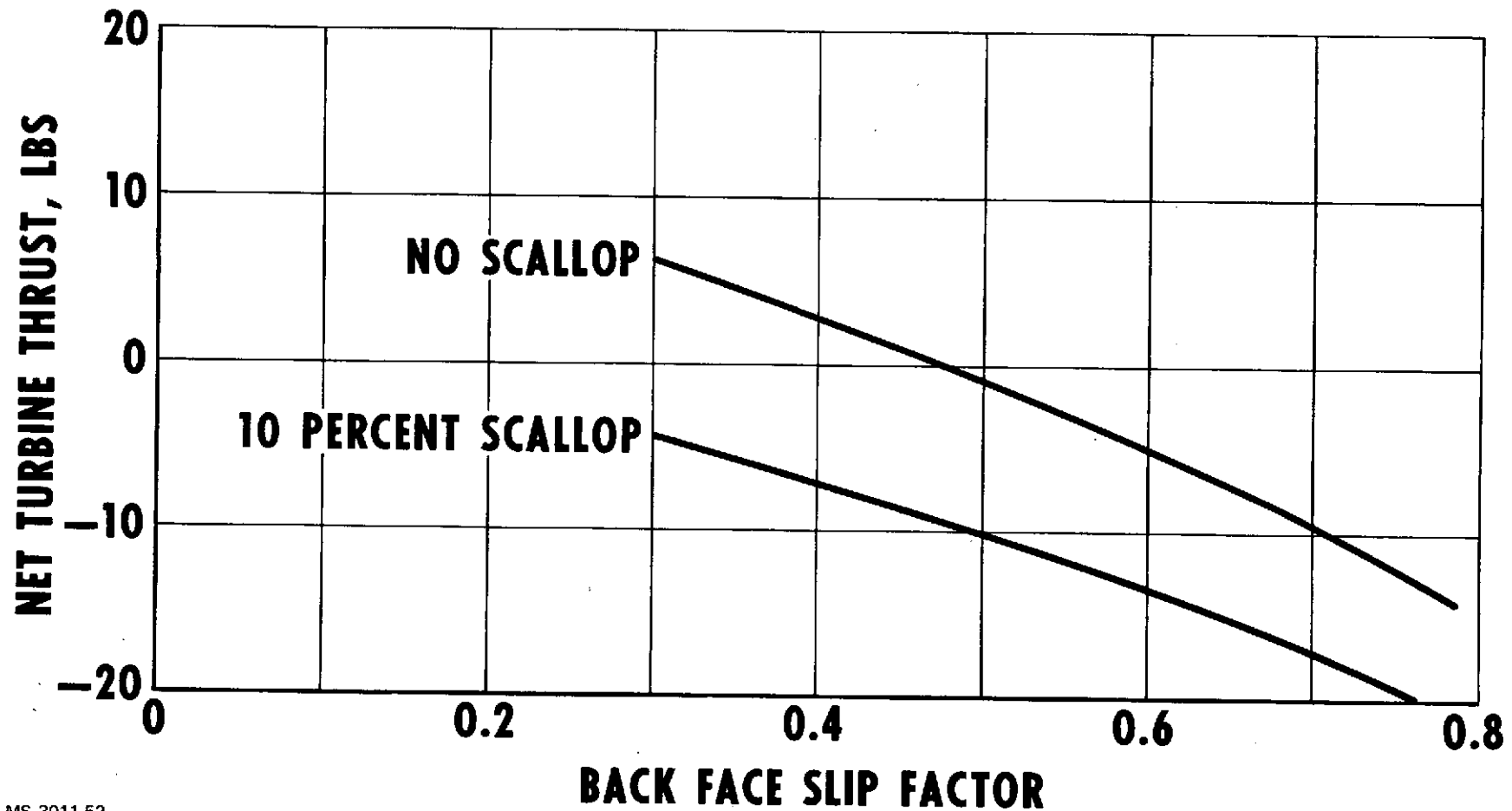
- JOURNAL DIAMETER = 1.00 IN.
- BEARING LENGTH = 0.90 IN.
- NUMBER OF FOILS = 8
- THICKNESS OF FOILS = 0.003 IN. + COATING
- RADIUS OF CURVATURE = 0.70 IN.
- SWAY SPACE = 0.006 IN.
- PREDICTED SPRING RATE = 3300 LB./IN.

MS 3011-22

TABLE 3-2



TURBINE THRUST CHARACTERISTICS

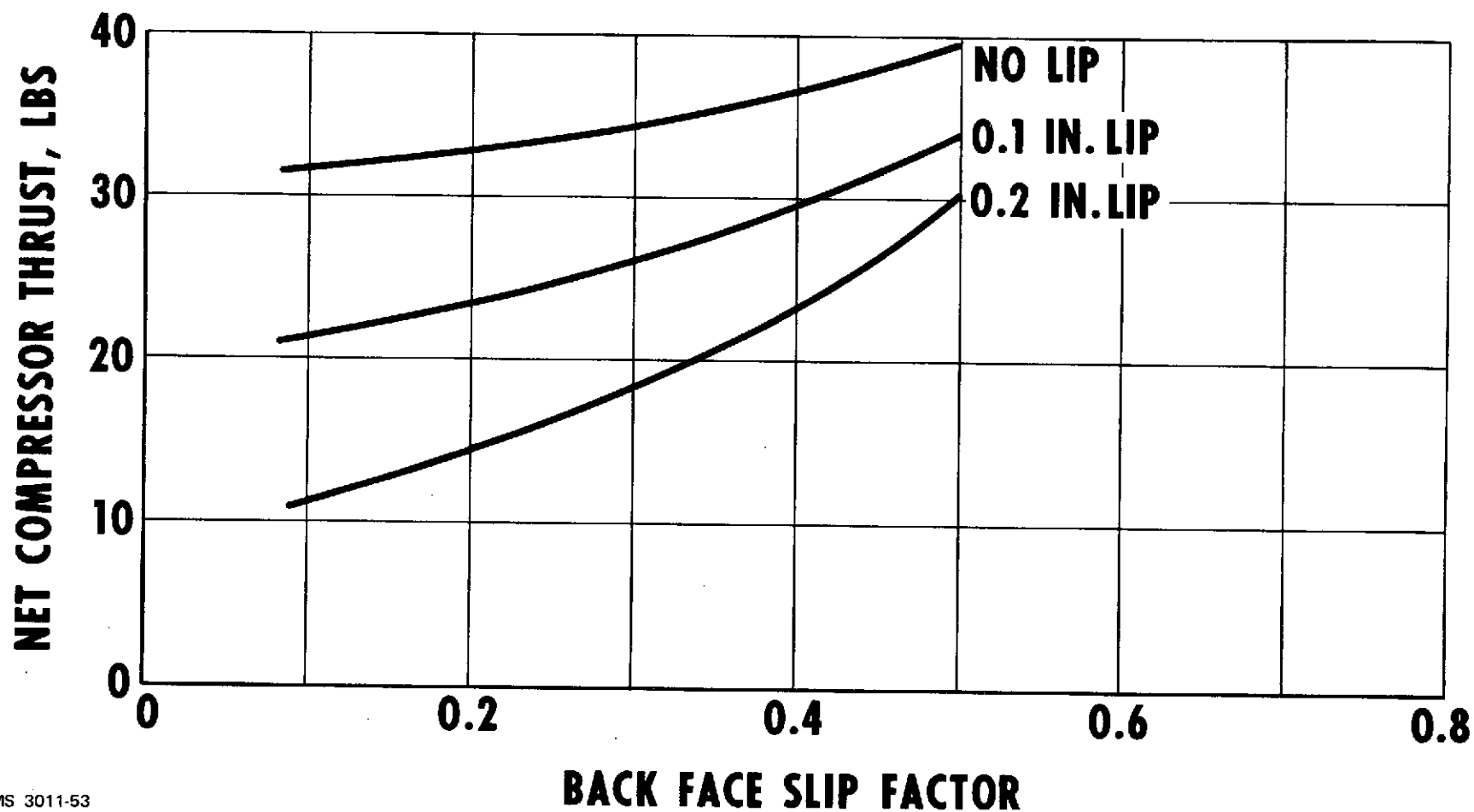


MS 3011-52

FIGURE 3-3



COMPRESSOR THRUST CHARACTERISTICS



MS 3011-53

FIGURE 3-4



THRUST BEARING DESIGN

- **OUTSIDE DIAMETER = 1.75 IN.**
- **INSIDE DIAMETER = 1.00 IN.**
- **NUMBER OF PADS = 10.00**
- **PAD THICKNESS = 0.003 IN. + COATING**
- **PAD PLATE THICKNESS = 0.005 IN.**
- **SPRING PLATE THICKNESS = 0.005 IN.**
- **PRE-LOAD = 3.50 LBS.**



ESTIMATED POWER LOSS SUMMARY

LOSSES IN WATTS

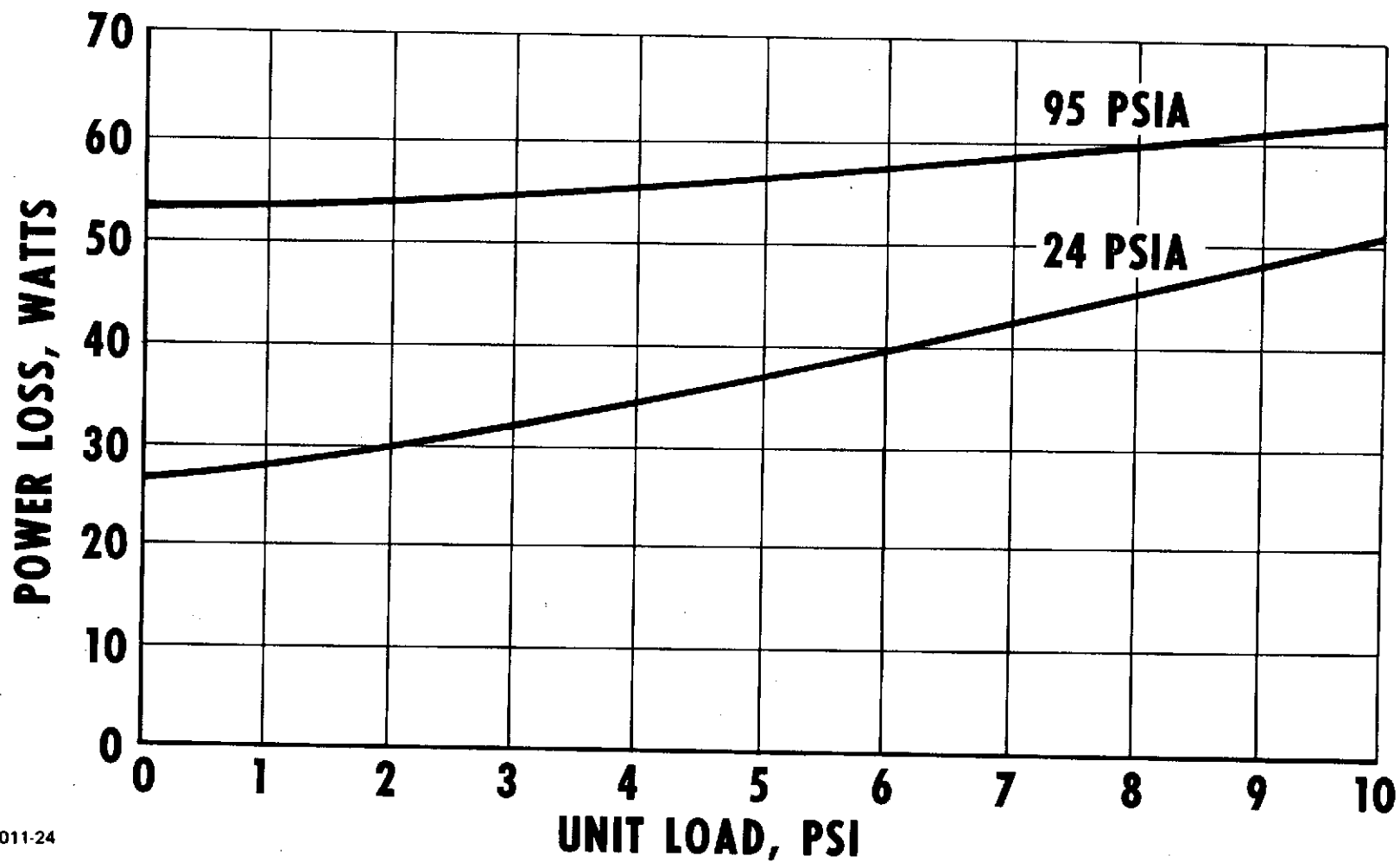
	24 PSIA	95 PSIA
JOURNAL BEARING (BOTH)	77	116
THRUST BEARING (LOADED SIDE)	59	74
THRUST BEARING (UNLOADED SIDE)	39	58
TOTAL	175	248

MS 3011-23

TABLE 3-4

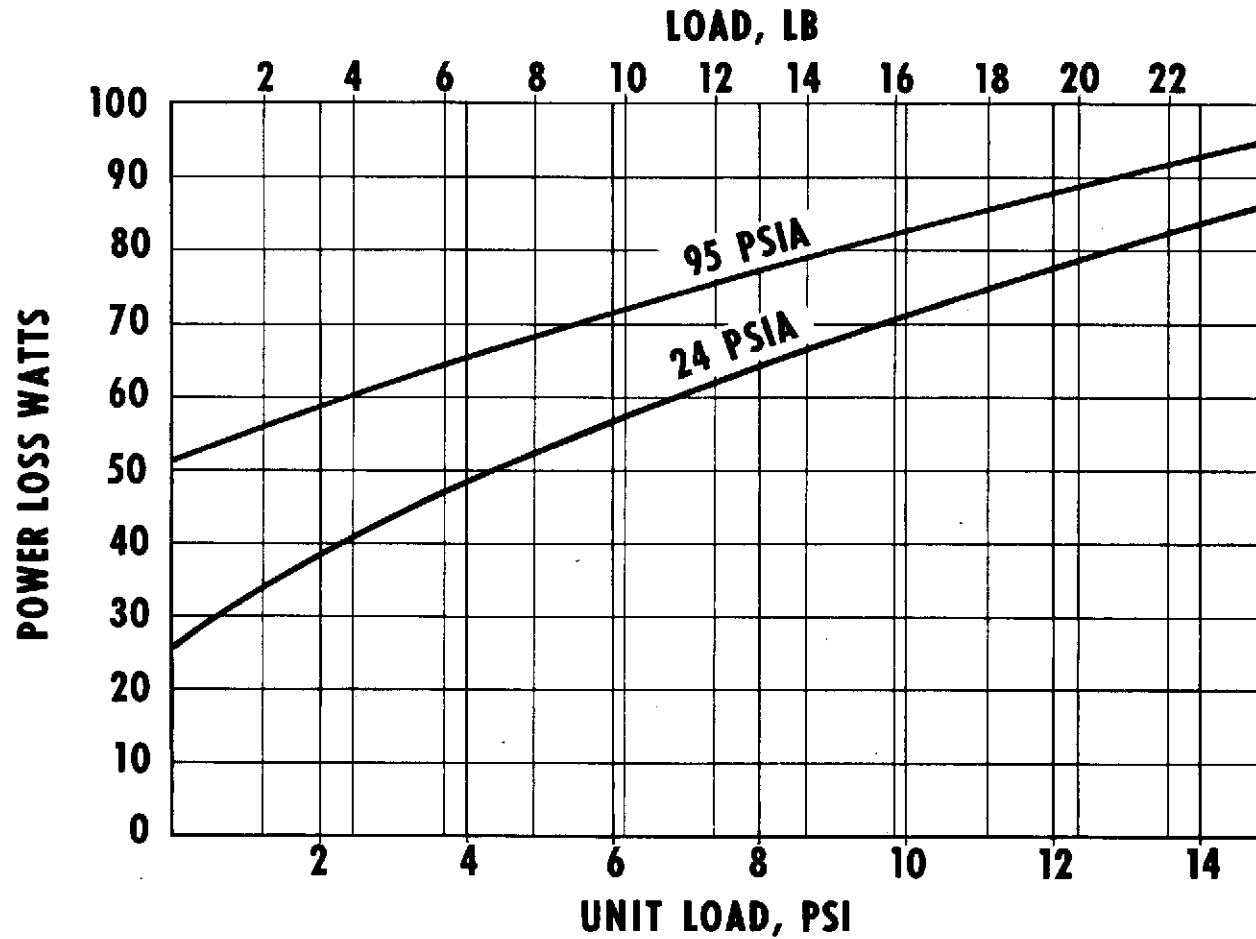


JOURNAL BEARING PERFORMANCE





THRUST BEARING PERFORMANCE



MS 3063-2



3.1.2 Mini-BRU Materials

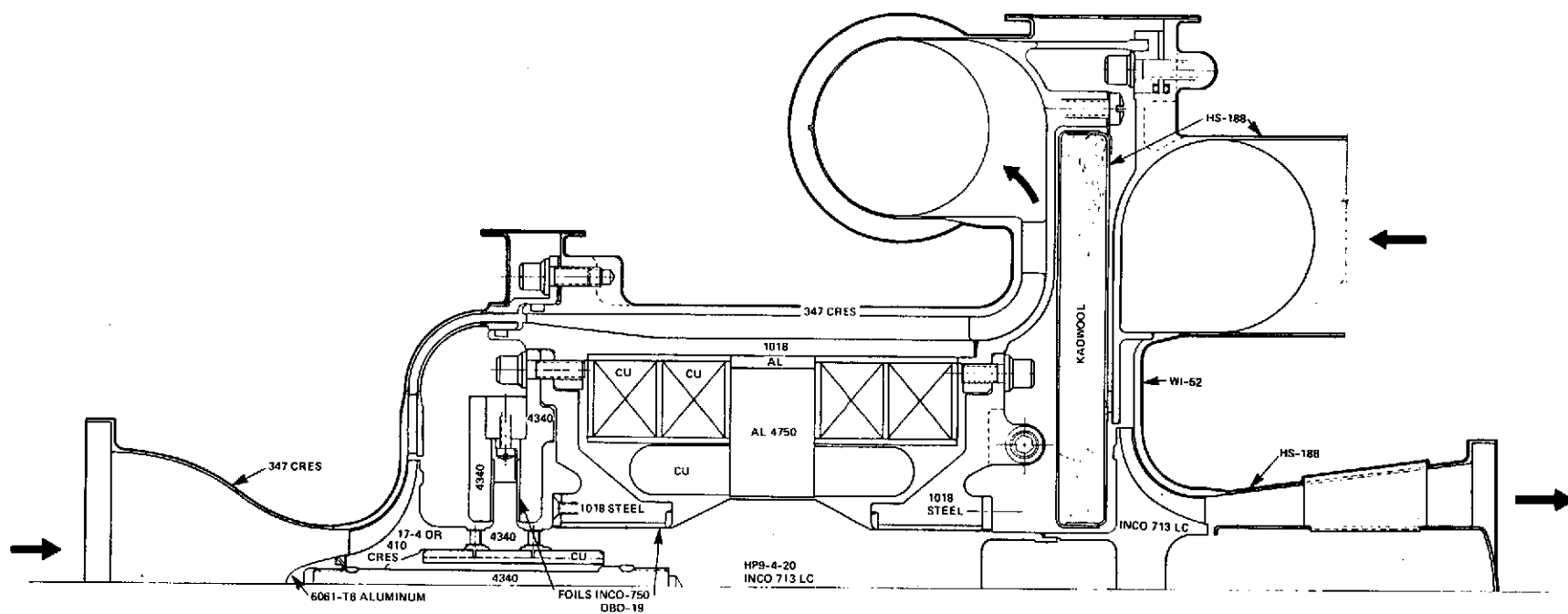
The materials for the mini-BRU were reviewed with tentative selections based on experience gained from the NASA BRU and TAC programs. Although the stresses are modest throughout the machine, the design life is at least 5 years. With the exception of the turbine-end of the machine, operating life up to 10 years can be expected.

Figure 3-5 shows the material selections for mini-BRU. Stainless steel will be used extensively in the center frame, compressor end, and alternator heat exchanger for corrosion resistance. 347 CRES was selected for its good fabricability and weldability. The compressor wheel will be 17-4 PH or 410 CRES, combining the high strength and good thermal conductivity necessary to extract heat from the thrust bearing. Bearing materials were selected to be consistent with the BRU and TAC designs. The alternator materials will result in good electromagnetic performance and thermal conductivity.

The turbine nozzle will be fabricated from cast WI-52 because of this metal's high temperature strength and stability and because it has good weldability with the HS-188 sheet metal used for the turbine scroll, shroud, and diffuser. The turbine wheel will be made from an Inco 713 LC casting. This selection is based on long experience in the design, fabrication and operation of many types of turbine wheels.



MINI-BRU MATERIALS



MS 3011-54

FIGURE 3-5



3.2 Aerodynamic Design

The aerodynamic design effort for this study focused on accurately assessing the performance of small, single stage radial compressors and turbines. Considerable attention was also given to coordinating the compressor diffuser design with the alternator heat exchanger design.

Table 3-5 summarizes the design factors which have an important effect on compressor efficiency.

Compressor pressure ratio, considered above, has direct effects on: the cycle flowrate required; the compressor size and efficiency; and the cycle pressure levels. The secondary effects of compressor pressure ratio selection include: rotating unit weight; heat exchanger/radiator heat transfer and pressure drop requirements; and cycle efficiency.

Evaluation of the combination of rotating and specific speeds selected for the aerodynamic components cannot be rigorously analyzed, excluding the remainder of the system. However, the combined effect of these two parameters can be viewed thusly: Any number of combinations of rotor and specific speeds can be found which result in a particular cycle pressure level and flowrate. With the system pressure levels and flowrates established, the system heat exchanger requirements are virtually invariant. The cycle optimization now centers on determining the particular combination of rotating speed and specific speed which results in the highest aerodynamic efficiency. High rotor speed results in small turbomachinery diameters and increasing losses due to increased blockage, surface losses and other size related phenomena. Low specific speeds eventually yield increasing losses due to increasing hub, shroud and end wall losses and clearance losses. After careful consideration of these factors and review of



COMPRESSOR DESIGN PROBLEMS AND SOLUTIONS

PROBLEM

RESULT

- LOW PHYSICAL SPEED IMPLIES LOW SPECIFIC SPEED

MODERATE EFFICIENCY PENALTY
BECAUSE OF NON-OPTIMUM SPECIFIC
SPEED

- SMALL SIZE

- (A) SMALL EXIT WIDTH + REASONABLE
RUNNING CLEARANCE

HIGH CLEARANCE LOSSES

TYPICAL $\frac{\text{CLEARANCE}}{\text{EXIT WIDTH}} = 0.030$

THIS CASE $\frac{\text{CLEARANCE}}{\text{EXIT WIDTH}} = 0.104$

- (B) TOLERANCES, FILLET RADII
RELATIVELY LARGER

SUBSTANTIAL EFFICIENCY PENALTY

- DESWIRL-RESWIRL VANES TO
ACCOMMODATE HEAT EXCHANGER

MODERATE EFFICIENCY PENALTY



NASA LeRC data (based on extensive testing of closed cycle machinery compressors) suitable performance penalties as a function of compressor size were defined as shown on the following page. A similar characteristic for the turbine is also shown.

The compressor design is summarized in Table 3-6 and Figure 3-6. The reference design is compared to existing component experience in Figure 3-7.

The turbine design is essentially a direct scale-down from existing single stage turbine. It is felt that there will be less difficulty in achieving the required turbine efficiency than will be the case for the compressor. The reference turbine design point is quite consistent with test data obtained on a number of radial turbines, as seen in Figure 3-8. The design point is summarized in Table 3-7 and Figure 3-9.

For reference purposes, the method of calculating inlet and mean specific speeds for the compressor and turbine is presented on Table 3-8.

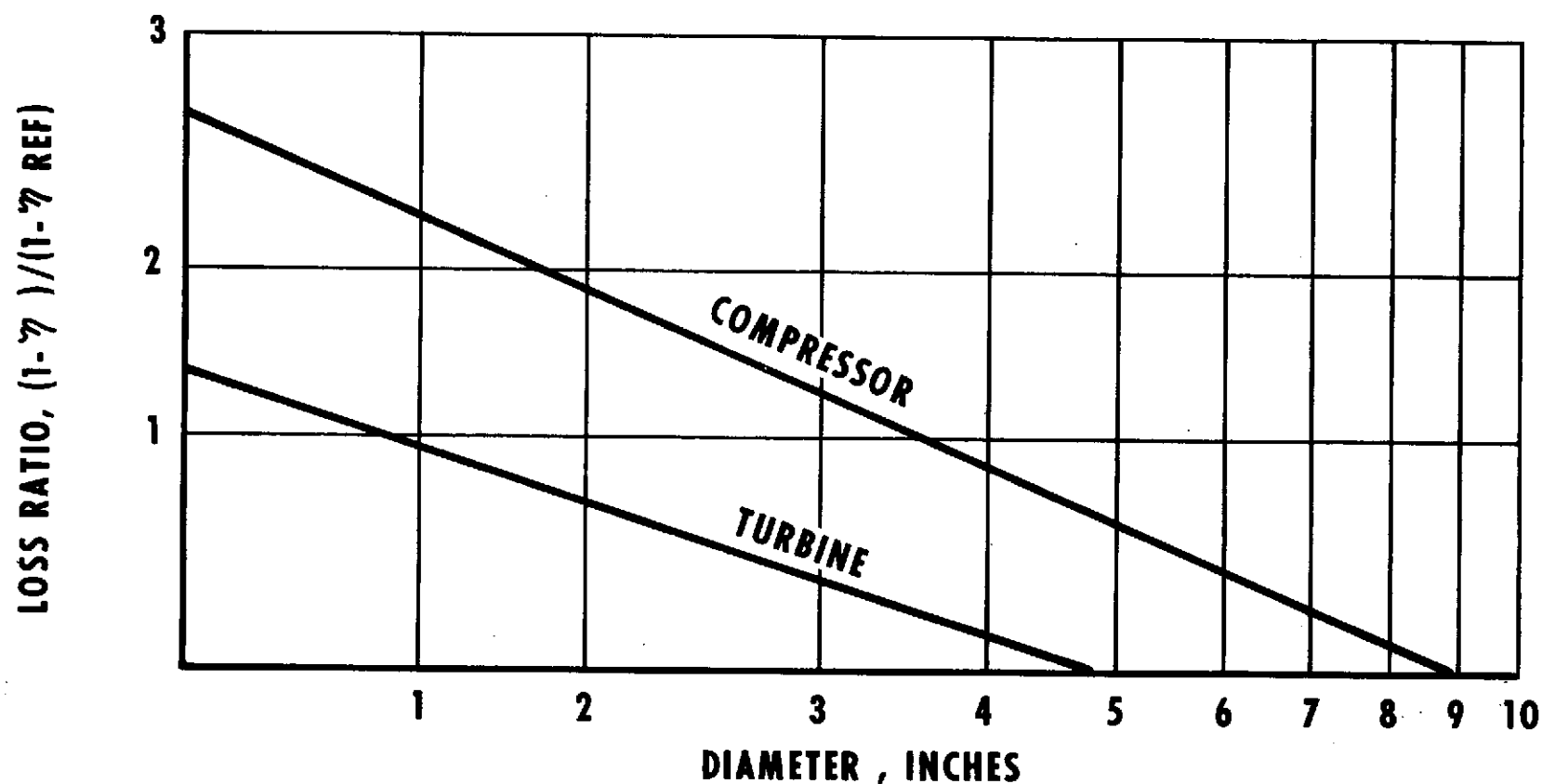
3.3 Alternator Electromagnetic Design

3.3.1 Design Selection

The alternator is a four-pole Rice machine designed to have high efficiency over a wide range of load conditions. Optimum system speed was determined to be 52,000 rpm. Preliminary studies at 41,000; 52,000; and 60,000 rpm for 2, 4, 6, and 8-pole alternator designs indicated that a four-pole machine had the smallest, lightest rotor at each speed (refer to Table 3-9). Although total alternator weight was slightly less for 6- and 8-poles, efficiency of these machines was reduced by higher windage losses caused, in turn, by larger diameter rotors.



COMPRESSOR AND TURBINE SIZE CORRECTION



MS 3063-6



COMPRESSOR DESIGN POINT CONDITIONS

PRESSURE RATIO

- TOTAL TO TOTAL = 1.50

EFFICIENCY

- TOTAL TO TOTAL = 0.76

SPEED

- SPECIFIC SPEED* = 44.3 (MEAN);
46.6 (INLET)
- RPM = 52,000

REYNOLDS NUMBER

$$\bullet \frac{\rho_{T1} U_2 D_2}{\mu_1} = 5.68 \times 10^6$$

INLET TOTAL PRESSURE

- $P_{IN} = 70.8$

INLET TOTAL TEMPERATURE

- $T_{IN} = 536^\circ R$

PERFORMANCE PARAMETERS

- CORRECTED FLOW = 0.0756 LB/SEC
- CORRECTED SPEED = 51,153 R.P.M.

MISCELLANEOUS

- AXIAL CLEARANCE = 0.008 IN.
- RADIAL CLEARANCE = 0.008 IN.
- NO. OF BLADES = 13
- VANED DIFFUSER
INLET GAS ANGLE = 76.7°
- VANED DIFFUSER
EXIT MACH NO. = 0.125

$$* \text{SPECIFIC SPEED} = (\text{RPM}) * (\text{VOLUMETRIC FLOW RATE})^{1/2} * \left(\frac{TR}{PR}\right)^{1/4} / (\Delta h_{isen})^{3/4}$$

MS 3063-4



COMPRESSOR VECTOR DIAGRAMS

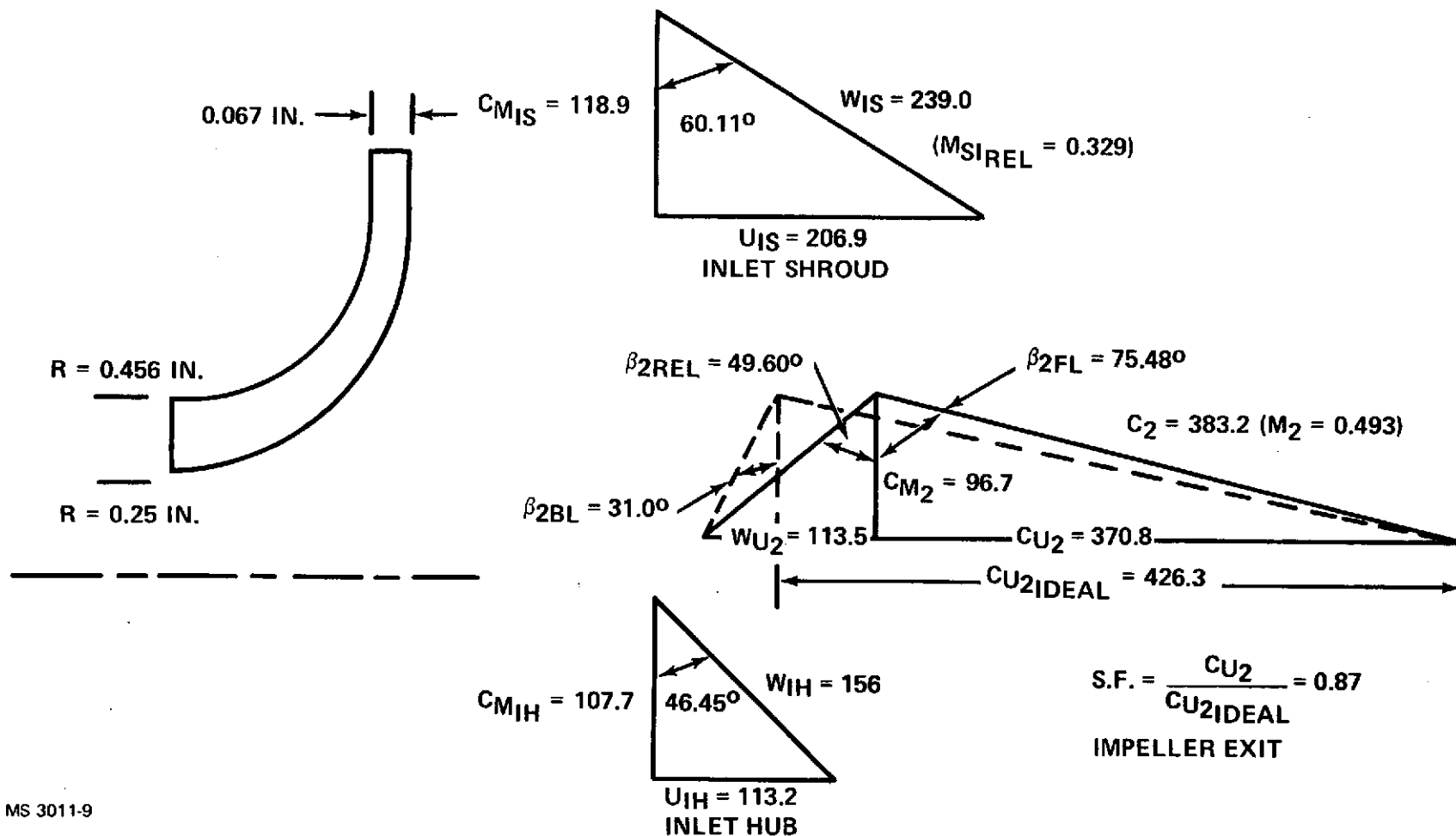
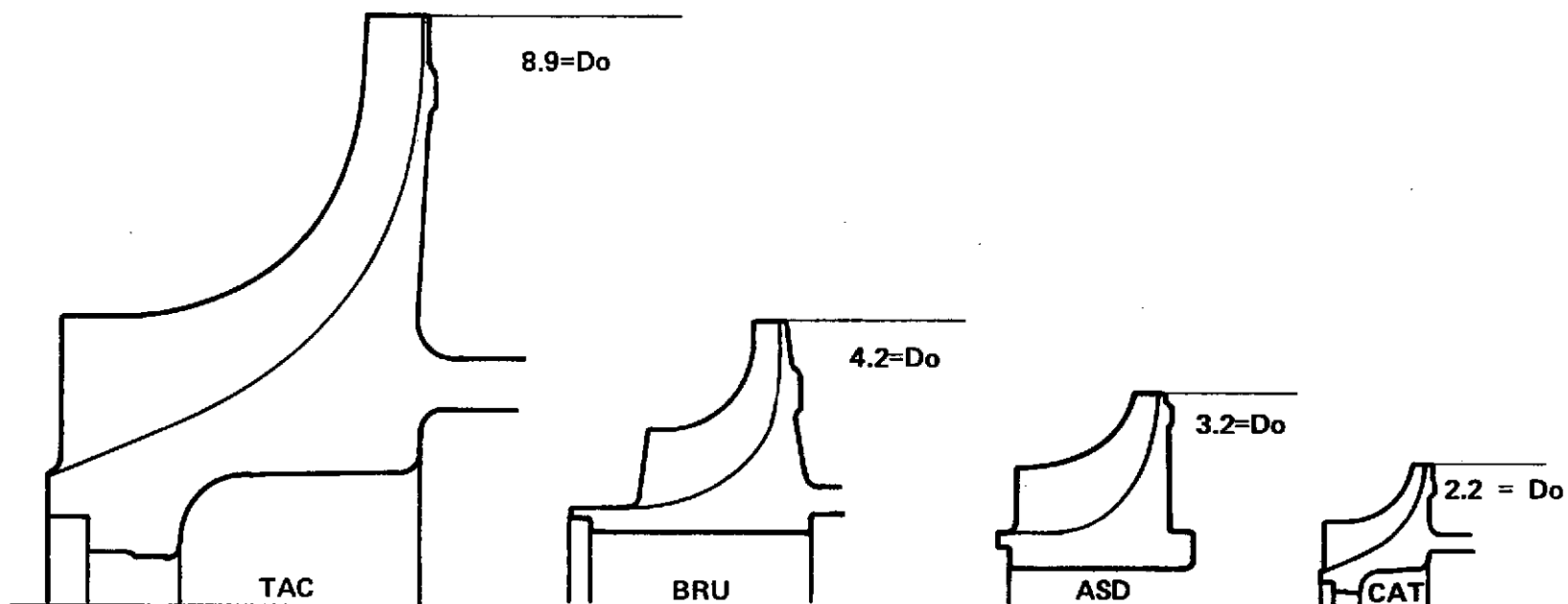


FIGURE 3-6



RADIAL COMPRESSOR EXPERIENCE



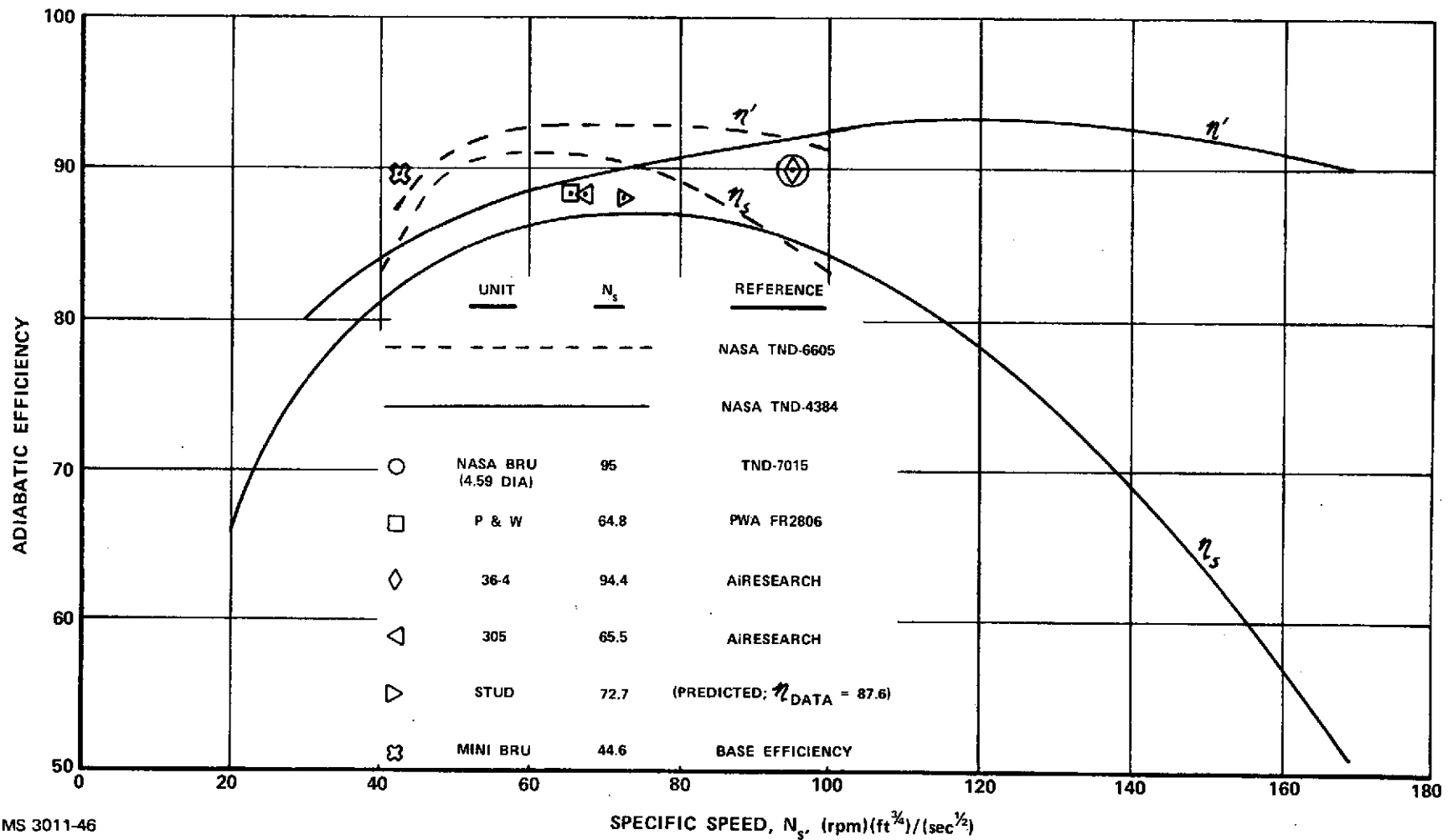
η_c	0.87	0.83	0.805	0.760
P_r	1.95	1.9	2.06	1.50
N	24,000	36,000	64,000	52,000
M	39.94	83.8	39.94	83.8
N_{SC}	0.086	0.105	0.10	0.070

MS 3011-8

FIGURE 3-7



TURBINE EFFICIENCY EXPERIENCE (1-12-73)



MS 3011-46

FIGURE 3-8



TURBINE DESIGN POINT CONDITIONS

PRESSURE RATIO

- TOTAL TO TOTAL = 1.47
- TOTAL TO D.E. STATIC = 1.4737

EFFICIENCY

- BASE TOTAL TO TOTAL = 0.896
- CLEARANCE LOSS = 0.039
- DISK FRICTION = 0.021
- NET ADIABATIC = 0.832

SPEED

- SPECIFIC SPEED* = 44.3 (MEAN);
41.5(INLET)
- RPM = 52,000.
- REYNOLDS NO. ($\dot{w}/\mu r$) = 64,500

INLET TEMPERATURE

- $T_{IN} = 2060^{\circ}R$

PERFORMANCE PARAMETERS * *

- EQUIV. FLOW = 0.0551 LB/SEC
- EQUIV. WORK = 9.903 BTU/LB
- EQUIV. TORQUE = 1.134 IN-LB
- EQUIV. SPEED = 42,899 RPM

MISCELLANEOUS

- WHEEL DIAMETER = 3.0 IN.
- PRESSURE RECOVERY = 0.6
- NOZZLE EXIT ANGLE = 72°
- DISCHARGE MACH NO. = 0.087
- AXIAL CLEARANCE = 0.008 IN.
- RADIAL CLEARANCE = 0.008 IN.
- NO. OF BLADES = 11

$$* \text{SPECIFIC SPEED} = \text{SPEED (RPM)} * (\text{VOLUMETRIC FLOW RATE})^{1/2} * \left(\frac{TR}{PR}\right)^{1/4} / (\Delta h_{isen})^{3/4}$$

** STANDARD CONDITIONS ASSUMED: $T=58.67^{\circ}F$; $P=14.7$ PSIA

MS 3063-3



TURBINE VECTOR DIAGRAMS

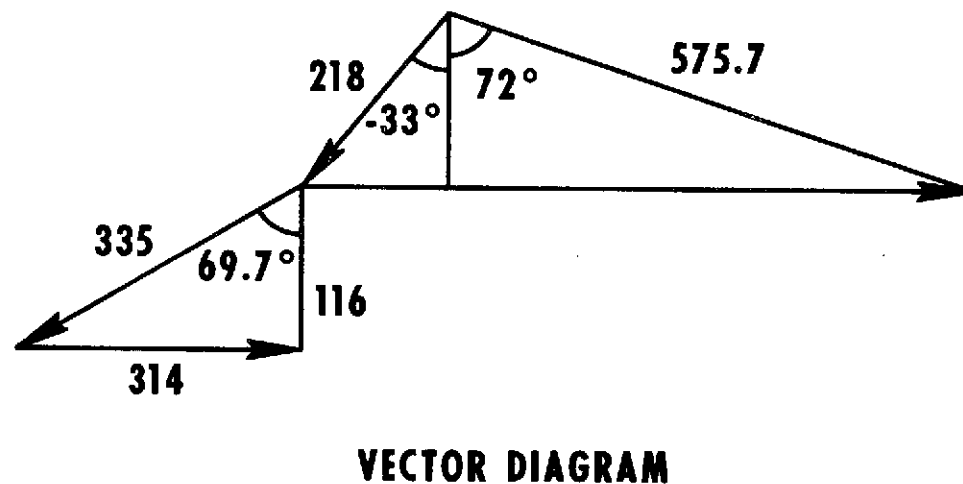
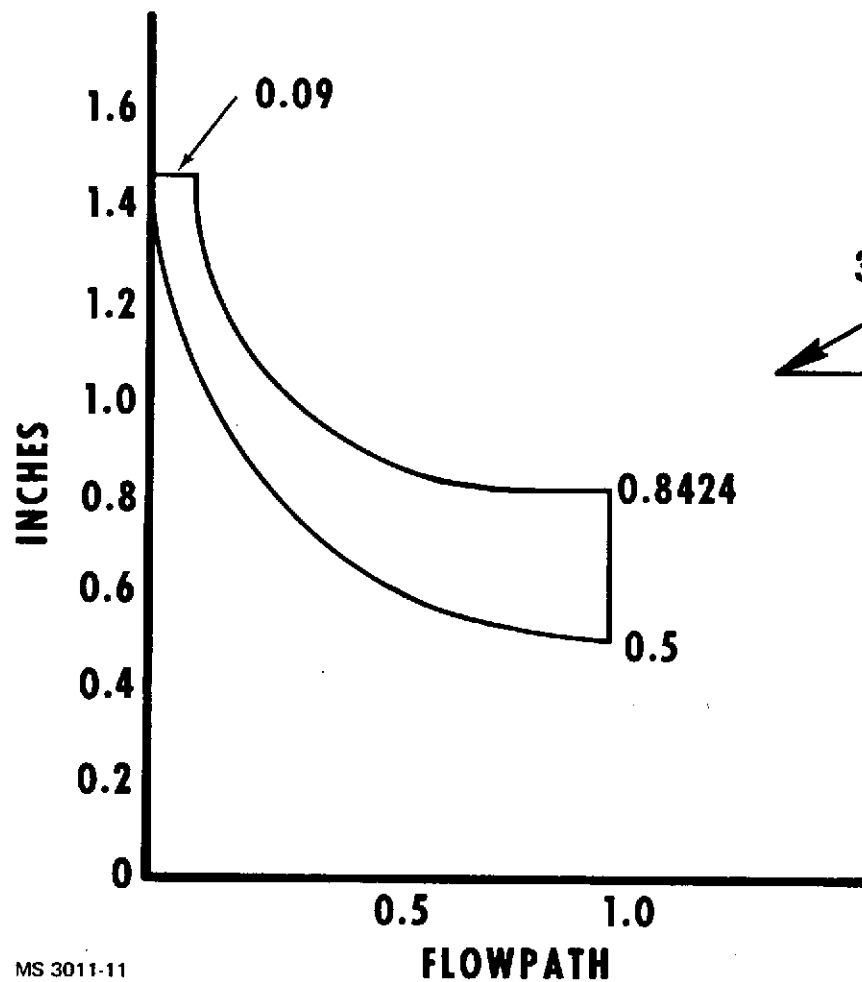


FIGURE 3-9



TABLE 3-8

METHOD OF CALCULATING SPECIFIC SPEED

$$\text{specific speed})_{\text{mean}} = \text{rpm} * (Q1D * Q2D)^{1/4} / (\Delta h)^{3/4}$$

$$Q2D = Q1D * PR/TR$$

$$Q1D = \frac{\text{°}}{\text{m}} \rho$$

$$\text{specific speed})_{\text{inlet}} = \text{specific speed})_{\text{mean}} * \left(\frac{TR}{PR} \right)^{1/4}$$

Compressor

$$\text{specific speed})_{\text{mean}} = 44.3$$

$$\text{specific speed})_{\text{inlet}} = 46.6$$

Turbine

$\rho =$

PR = 1.47

P_{IN} = 105 psia

$\frac{\text{°}}{\text{m}} =$

ΔT = 244.8

T_{IN} = 2060

TR = 1.136

$$\text{specific speed})_{\text{inlet}} = \frac{52,000 * (0.875)^{1/2}}{(0.0592 * 244.8 * 778.16)^{3/4}} =$$

$$\frac{52,000}{1,100} * (0.875)^{1/2}$$

$$\frac{52,000}{1,100} * 0.935 = 44.3$$

$$\text{specific speed})_{\text{mean}} = \text{specific speed})_{\text{inlet}} * \left(\frac{PR}{TR} \right)^{1/4}$$

$$44.3 * \left(\frac{1.136}{1.470} \right)^{1/4} = 41.5$$



MINI-BRU SPEED AND POLE STUDY

DATA DESIGN POINT: 2.169 KW, 66.24 VOLTS, 13.19 AMPS -0.8271 PF, $X_L = 0.15$ PU, $Z_{BASE} = 5.02$ OHMS								LOSSES - WATTS						
	SPEED RPM	FREQ Hz	N* %	N _e %	DROT INCHES	WTOT POUNDS	WROT POUNDS	STATOR COPPER	STACK	POLE HEAD	FIELD	STRAY	FRICTION & WINDAGE*	TOTAL
8 POLE	41,000	2733	86.36	90.10	1.9	7.060	1.828	46.25	81.76	8.97	97.61	3.81	104.0	342.5
	52,000	3467	84.25	89.68	1.8	5.937	1.532	40.22	101.1	8.50	94.7	5.02	156.0	405.5
	60,000	4000	82.84	88.96	1.7	5.433	1.406	36.39	125.2	7.58	92.67	5.66	179.2	446.8
6 POLE	41,000	2050	87.99	90.95	1.8	6.750	1.509	52.99	59.29	17.20	83.83	2.65	80.08	296.0
	52,000	2600	86.42	90.71	1.7	5.714	1.278	45.71	76.37	16.20	80.78	2.94	118.80	340.8
	60,000	3000	85.66	90.47	1.6	5.170	1.150	41.25	91.20	14.35	78.46	3.24	134.40	363.0
4 POLE DESIGN SELECTION	41,000	1367	88.41	90.67	1.7	7.090	1.280	64.09	41.04	45.59	71.00	1.40	61.18	284.3
	52,000	1733	87.51	90.78	1.6	5.977	1.079	55.33	52.50	42.48	68.33	1.57	89.35	309.5
	60,000	2000	87.26	90.89	1.5	5.366	0.956	50.05	61.96	37.30	66.35	1.76	99.29	316.7
2 POLE	41,000	683	87.83	90.77	1.8	11.03	1.451	96.52	27.02	56.93	39.05	0.98	80.0	300.5
	52,000	867	86.88	91.21	1.7	9.183	1.204	83.41	33.83	53.68	36.82	1.08	118.4	327.2
	60,000	1000	86.66	91.55	1.6	8.227	1.068	75.71	38.77	48.13	36.12	1.21	133.7	333.7

*INCLUDES WINDAGE LOSSES IN THE MAIN GAP AND 1/2 OF THE CONE SECTION

MS 3011-33

TABLE 3-9



Since high efficiency rather than minimum weight was a design goal, the four-pole machine was selected for further optimization and analysis. Practical considerations of fabrication also favored a four-pole machine.

The analysis employed a math model based on conventional salient-pole alternator theory. The model included the following routines:

- (a) Pole and flux collector geometry; stator geometry.
- (b) Pole leakage
- (c) Rotor windage loss
- (d) End bell design and leakage
- (e) Control and series field design
- (f) Pole head loss
- (g) Eddy factor for stator windings
- (h) Apparent power factor, a-c volts and amps, d-c volts and amps, distortion factor, and rectifier losses for operation into a three-phase, full-wave rectifier load.
- (i) Iteration of stator turns to achieve specified commutation reactance.

The operation of the alternator math model is such that the initial or "design" load condition synthesizes a basic geometry. Then, when subsequent load conditions are applied, performance is calculated using the initial basic geometry. The basic geometry can readily be changed either by changing the initial "design" load data or by changing initial design parameters such as pole flux density, winding reactance, or an air gap dimension.



Figure 3-10 describes the magnetic circuit used in the alternator math model. An understanding of this circuit is necessary to interpret the magnetic data presented in Table 3-10.

The math model made it possible to rapidly compare many design variations while maintaining consistent values for flux densities, winding reactance, or geometric features as appropriate. Figure 3-11 shows the effect of different designs on efficiency at various load conditions. The 1.5 in. diameter rotor design was selected to meet the maximum output requirements and provide good efficiency at part load. The design point rating of the selected machine is:

- o Four pole Rice
- o 65.9 volts, L-N
- o 12.27 amps per phase
- o 2.0 kw
- o 2.43 kva
- o 3 phase "Y" connected
- o 52,000 rpm
- o 1733 Hz

A summary of performance of the recommended design for d-c and a-c loads is presented in the following section.

3.3.2 Design Analysis

Table 3-10 presents magnetic circuit data for the initializing or design load case and for three different d-c load conditions corresponding to the power available from 1, 2, and 3 energy capsules.

Figure 3-12 presents the alternator wiring diagram used in this analysis and clarifies the proposed excitation system.



RICE ALTERNATOR MAGNETIC CIRCUIT

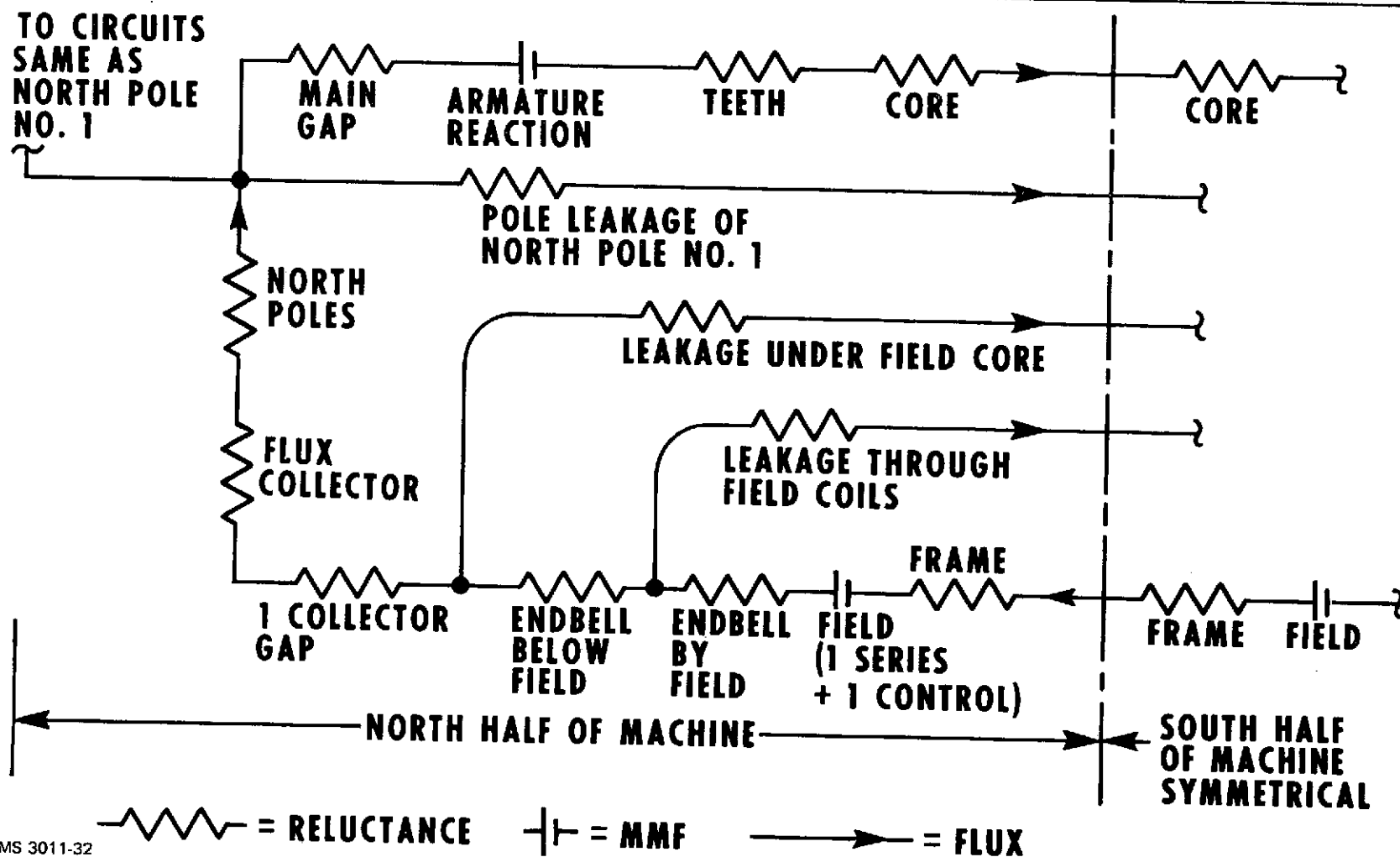


FIGURE 3-10



MAGNETIC DATA

	PERMEANCE LINES/AT	MAGNETIC LENGTH, IN.	AREA, IN ²	INPUT POWER											
				0.883 KW (1 CAPSULE)			1.805 KW (2 CAPSULES)			2.16 KW (DESIGN)			2.574 KW (3 CAPSULES)		
				KILOLINES PER POLE	KILOLINES PER IN ²	MMF/POLE, AMP TURNS	KILOLINES PER POLE	KILOLINES PER IN ²	MMF/POLE, AMP TURNS	KILOLINES PER POLE	KILOLINES PER IN ²	MMF/POLE, AMP TURNS	KILOLINES PER POLE	KILOLINES PER IN ²	MMF/POLE, AMP TURNS
MAIN GAP TEETH CORE ARMATURE REACTION POLE LEAKAGE	27.94	0.0205	0.4317	22.10 8.195	36.85 81.90 25.60	236.0 4.786 0.2278 52.53	24.64 11.55	41.08 91.31 28.54	263.1 7.624 0.2464 142.4	25.55 12.97	42.61 94.70 29.60	272.9 11.72 0.2559 179.6	26.70 27.94	44.52 98.96 30.93	285.0 30.3 0.260 222.0
POLES*	9.73 10.48	0.0200	0.8111	60.59	74.70	79.20	72.38	89.23	127.6	77.06	95.00	165.5	83.45	102.9	277.7
FLUX COLLECTOR*			0.7823	60.59	74.45	26.29	72.38	92.52	43.31	77.06	98.50	62.51	83.45	106.7	154.1
COLLECTOR GAP*			3.185	60.59	19.02	119.1	72.38	22.73	142.3	77.06	24.19	151.5	83.45	26.2	164.0
FIELD LEAKAGE*				5.05			7.07			8.21			11.04		
UNDER COIL*				5.47			7.68			8.92			12.01		
THRU COIL*			1.066	65.64	61.58	4.198	79.45	74.54	6.339	85.27	80.00	7.543	94.47	88.65	11.48
END BELL*			1.177	71.11	60.39	4.234	87.13	74.00	6.522	94.19	80.00	7.090	106.5	90.41	13.10
FRAME			22.35	71.11	60.39	9.580	87.13	74.00	14.76	94.19	80.00	17.89	106.5	90.44	29.63
TOTAL AMP TURNS/1/2 MACHINE						536.0			754.2			877.2			1188.

*VALUES PER 1/2 MACHINE (NOT PER POLE) IN ACCORDANCE WITH MAGNETIC CIRCUIT



ALTERNATOR EFFICIENCY DATA

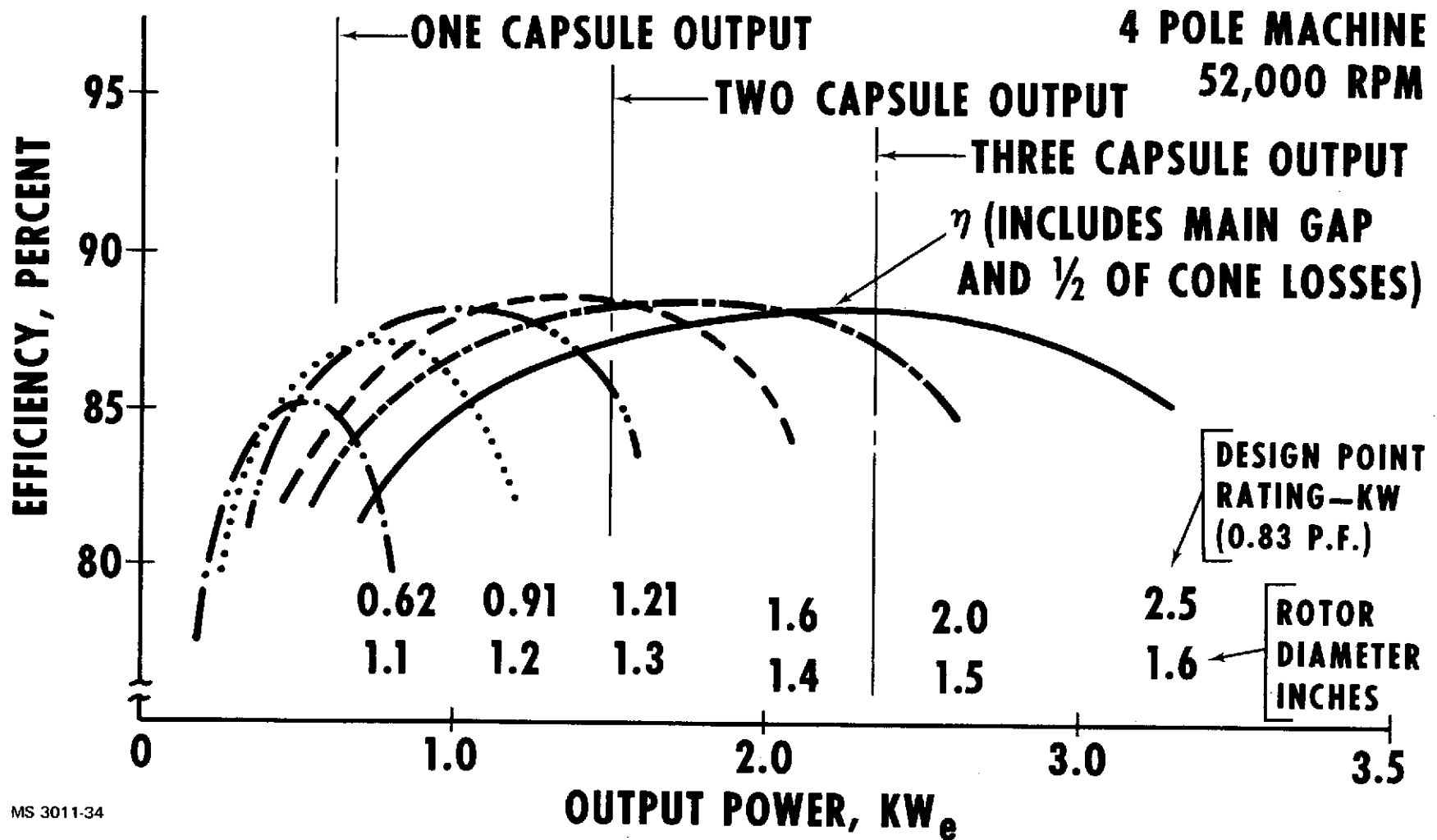
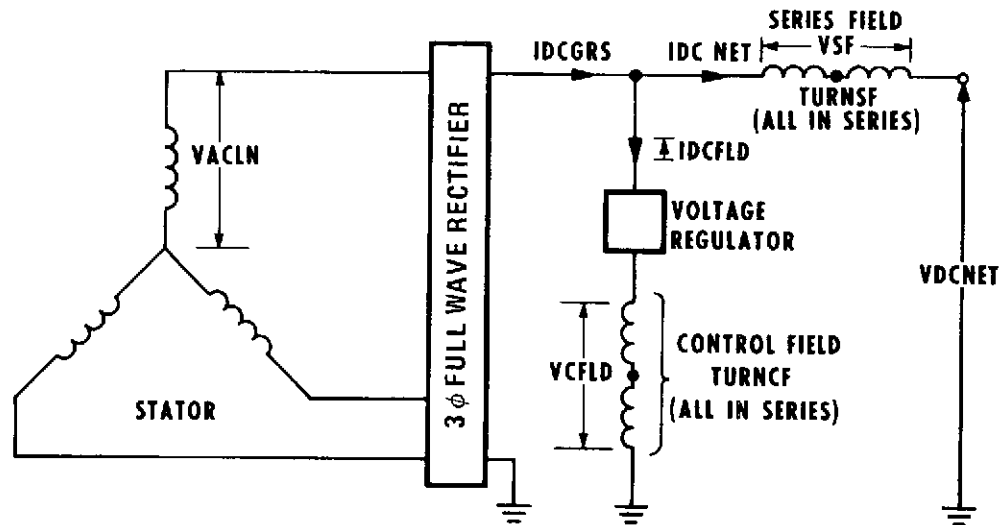


FIGURE 3-11



WIRING DIAGRAM



- IDCGRS** - TOTAL DC CURRENT FROM RECTIFIER, AMPS
- IDCFLD** - CONTROL FIELD CURRENT, AMPS
- IDCNET** - DC LOAD CURRENT, AMPS
- VACLN** - LINE TO NEUTRAL VOLTAGE, AC VOLTS
- VCFLD** - VOLTAGE ACROSS CONTROL FIELD, DC VOLTS
- VSF** - VOLTAGE ACROSS SERIES FIELD, DC VOLTS

- VDC NET** - DC BUS VOLTAGE, DC VOLTS
- TURN SF** - TOTAL NUMBER OF TURNS IN SERIES FIELD
- TURN CF** - TOTAL NUMBER OF TURNS IN CONTROL FIELD

MS 3011-29

FIGURE 3-12



Significant electrical parameters of the alternator are listed in Table 3-11. These values are based on the Design Load case in which base alternator voltage is 65.9 V_{L-N}; base alternator current is 12.27 amps ac, and the frequency is 1733 Hz. Since a rectifier load is specified, the subtransient reactance is a critical parameter for this application because it determines the apparent power factor of the load as a function of load current.

When the alternator is regulated to maintain a constant rectified load voltage of 120 vdc, change in load current will cause a change in apparent load power factor, alternator ac voltage and field excitation. This is depicted in Figure 3-13.

Operation at 3 pu short circuit is specified; therefore, a series field is required to provide adequate excitation for the short circuit condition. This requirement defines the slope of the series field excitation line shown in Figure 3-14. The control field curve is simply the difference between the total field curve and the series field curve. The use of a series field is not conducive to optimum efficiency at every load condition since the current density is not equal in the series and control field windings, except at certain points.

The losses in the control field and series field, the rectifier loss, and the alternator efficiency are presented in Table 3-12 and Figure 3-15.

Alternator losses at unity power factor at various ac load conditions are tabulated in Table 3-13. The corresponding efficiency and loss curves are shown in Figure 3-16.

Alternator characteristics at unity pf load conditions, no-load, and short circuit are shown in Figure 3-17. Zero pf characteristics are shown in Figure 3-18.



ALTERNATOR PARAMETERS

REACTANCE & RESISTANCE, FIELD TIME CONSTANT

BASE IMPEDANCE	Z _{BASE} 5.371 OHM
RESISTANCES ARMATURE (@ 360°F) FIELD (@ 350°F) – 2 CONTROL FIELDS IN SERIES – 2 SERIES FIELDS IN SERIES	R _A 0.0944 OHM R _{CF} 9.295 OHMS R _{SF} 0.0669 OHMS
REACTANCES DIRECT AXIS SYNCHRONOUS QUADRATURE AXIS SYNCHRONOUS ARMATURE LEAKAGE FIELD LEAKAGE ZERO SEQUENCE NEGATIVE SEQUENCE TRANSIENT SUBTRANSIENT DIRECT* SUBTRANSIENT QUADRATURE*	X _D 1.040 PER UNIT X _Q 0.556 PER UNIT X _L 0.152 PER UNIT X _F 0.286 PER UNIT X ₀ 0.066 PER UNIT X ₂ 0.200 PER UNIT X _{DU} 0.439 PER UNIT ≈X'' _D 0.382 PER UNIT ≈X'' _Q 0.382 PER UNIT
FIELD TIME CONSTANT (HOT) SHORT CCT OPEN CCT	TPD 0.0137 SECONDS 0.0369 SECONDS

*ESTIMATED VALUE FOR RICE ROTOR WITH NO DAMPER CAGE

MS 3063-14

TABLE 3-11



EXCITATION CHARACTERISTICS — DC LOAD

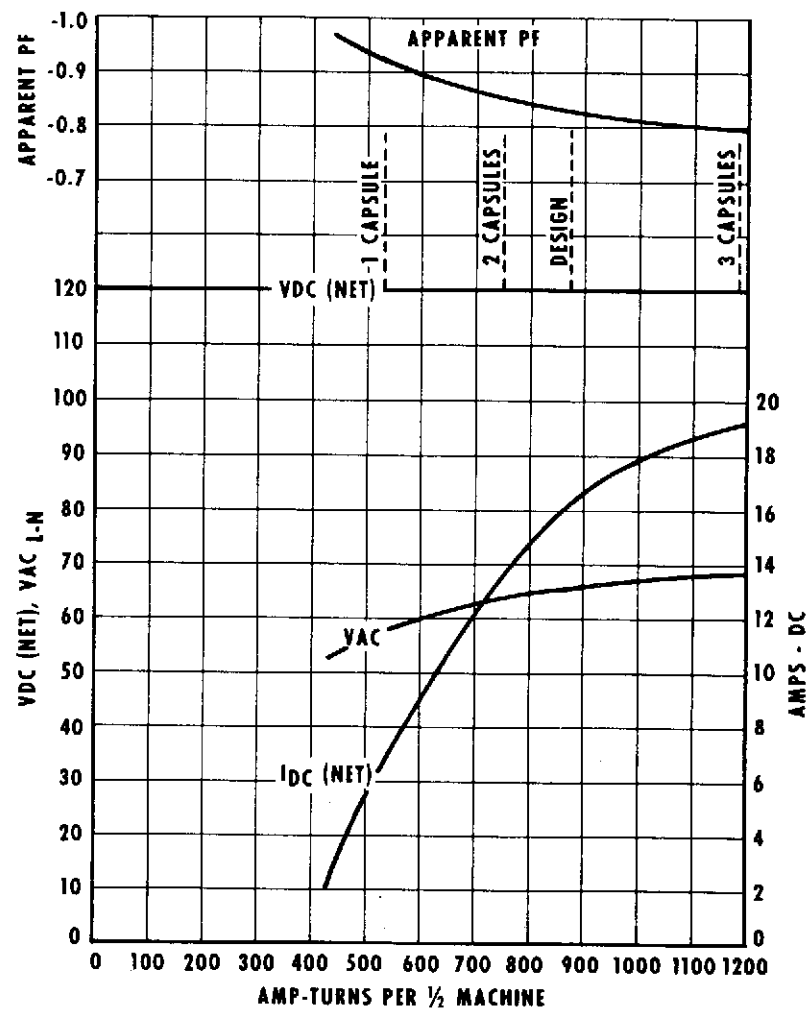
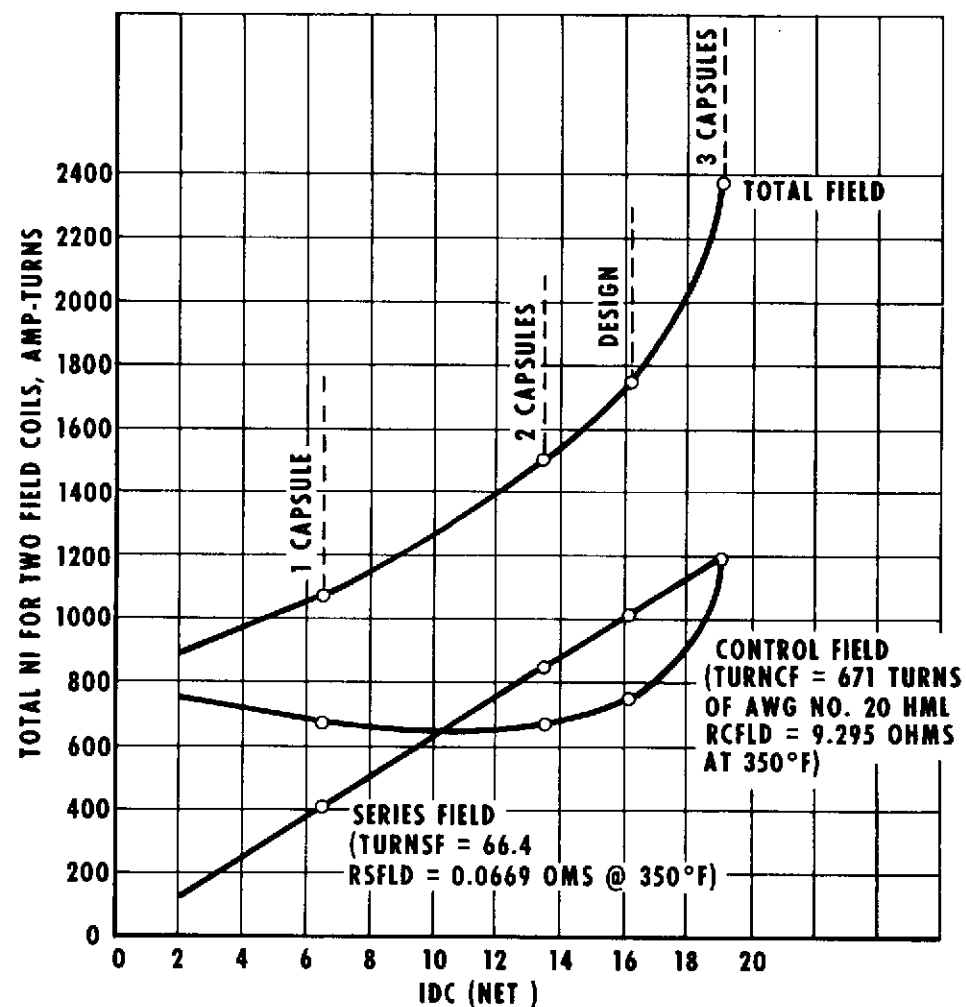


FIGURE 3-13



FIELD CURRENT – DC LOAD



MS 3063-16

FIGURE 3-14



ALTERNATOR AND RECTIFIER LOSSES (KW)

	INPUT POWER					
	2 AMP DC NET (0.315 KW)	4 AMP DC NET (0.563 KW)	1 CAPSULE (0.883 KW)	2 CAPSULES (1.805 KW)	DESIGN (2.16 KW)	3 CAPSULES (2.574 KW)
CORE	0.00984	0.01050	0.01145	0.01423	0.01530	0.01671
TEETH	0.01709	0.01823	0.01988	0.02471	0.02657	0.02902
STATOR COPPER	0.00086	0.00301	0.00742	0.03025	0.04278	0.05948
STRAY	0.00004	0.00012	0.00030	0.00124	0.00175	0.00244
POLE HEAD	0.01418	0.01565	0.01809	0.02780	0.03249	0.03871
CONTROL FIELD	0.01153	0.01031	0.00921	0.00914	0.01406	0.02879
SERIES FIELD	0.00024	0.00095	0.00247	0.01077	0.01550	0.02163
WINDAGE	0.01281	0.02071	0.02962	0.05234	0.06019	0.06904
TOTAL (ALTERNATOR)	0.06659	0.07948	0.09844	0.17048	0.20864	0.26582
EFFICIENCY (ALT)	79.8%	86.0%	88.9%	90.6%	90.3%	89.7%
RECTIFIER (BASED ON LOSS EQUIVALENT TO 1.5 V _{FWD})	0.00330	0.00625	0.00992	0.02047	0.02450	0.02908
TOTAL LOSSES	0.06989	0.08573	0.10836	0.19095	0.23314	0.2949
EFFICIENCY (DC NET)	77.5%	84.8%	87.7%	89.4%	89.3%	88.6%

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TABLE 3-12



EFFICIENCY AND LOSSES — DC LOAD

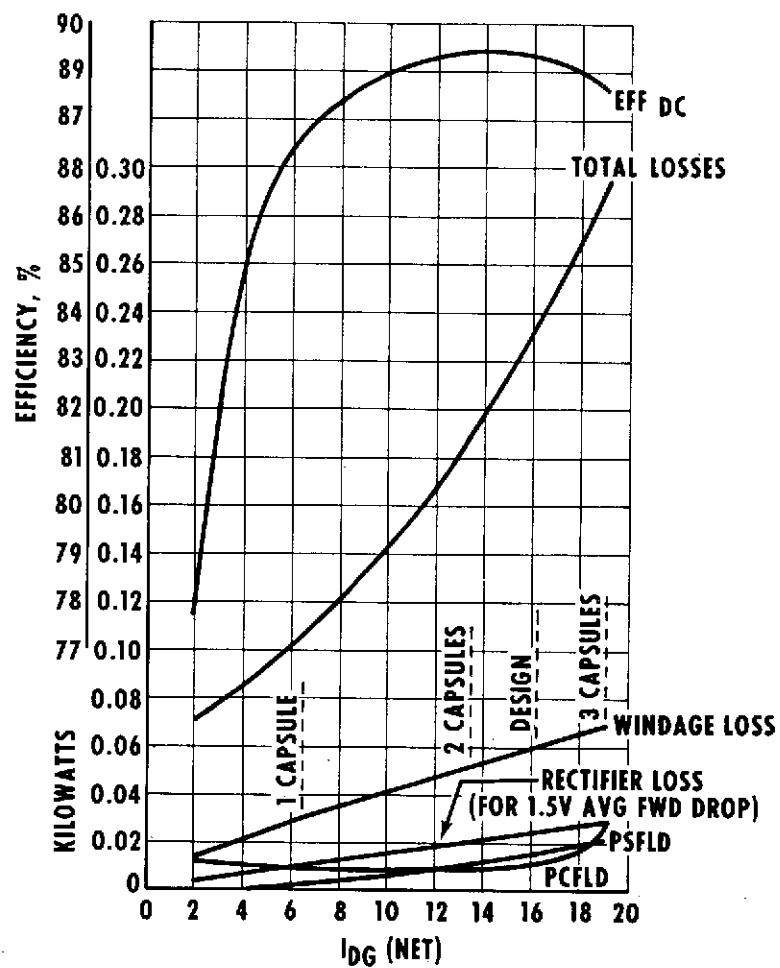


FIGURE 3-15



ALTERNATOR LOSSES (KW) AT UNITY P.F.

AC MODE: $I_{AC} = 12.27$ AMPS $V_{AC} = 65.9$ VOLTS

PER UNIT CURRENT	LOAD CONDITIONS					
	0	0.25	0.50	0.75	1.00	1.25
CORE	0.01471	0.01484	0.01485	0.01482	0.01479	0.01483
TEETH	0.02555	0.02577	0.02579	0.02573	0.02569	0.02575
STATOR COPPER	0.0000	0.00267	0.01070	0.02407	0.04278	0.06685
STRAY	0.0000	0.00011	0.00044	0.00099	0.00175	0.00274
POLE HEAD	0.02089	0.02175	0.02379	0.02709	0.03177	0.03788
FIELD*	0.01055	0.01125	0.01284	0.01536	0.01911	0.02417
WINDAGE	0.000	0.02376	0.04067	0.05589	0.07019	0.08386
TOTAL	0.07170	0.1002	0.1291	0.1639	0.2061	0.2561
EFFICIENCY		85.81%	90.37%	91.73%	92.16%	92.21%

*BASED ON DESIGN WITHOUT SERIES FIELD COMPOUNDING

MS 3063-19

TABLE 3-13



EFFICIENCY & LOSSES – UNITY POWER FACTOR LOAD

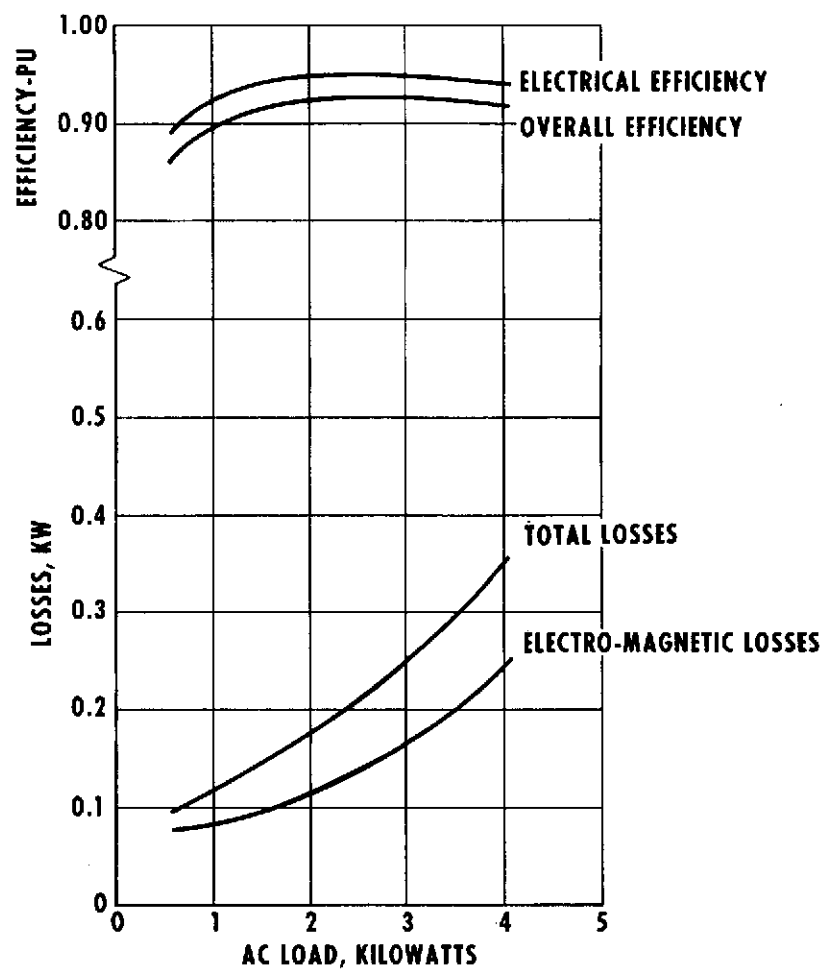


FIGURE 3-16



EXCITATION CHARACTERISTICS NO LOAD AND SHORT CIRCUIT

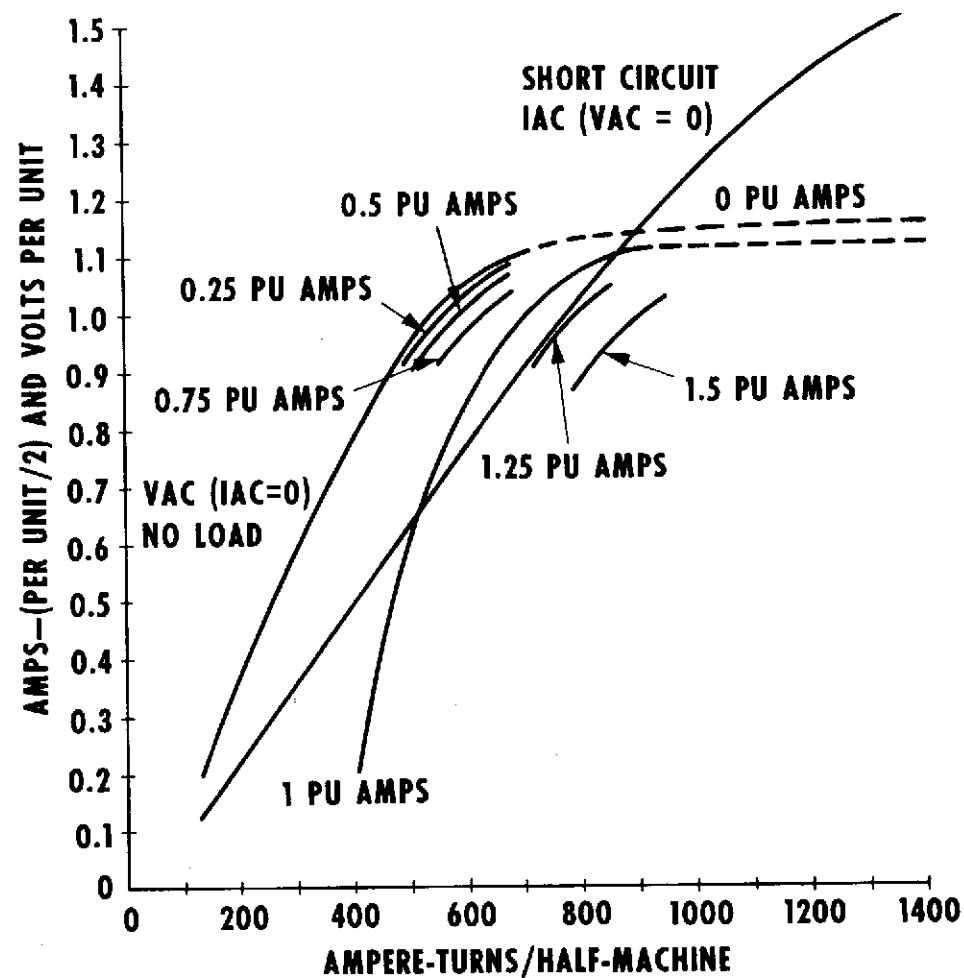


FIGURE 3-17



ALTERNATOR CHARACTERISTICS AT ZERO POWER FACTOR

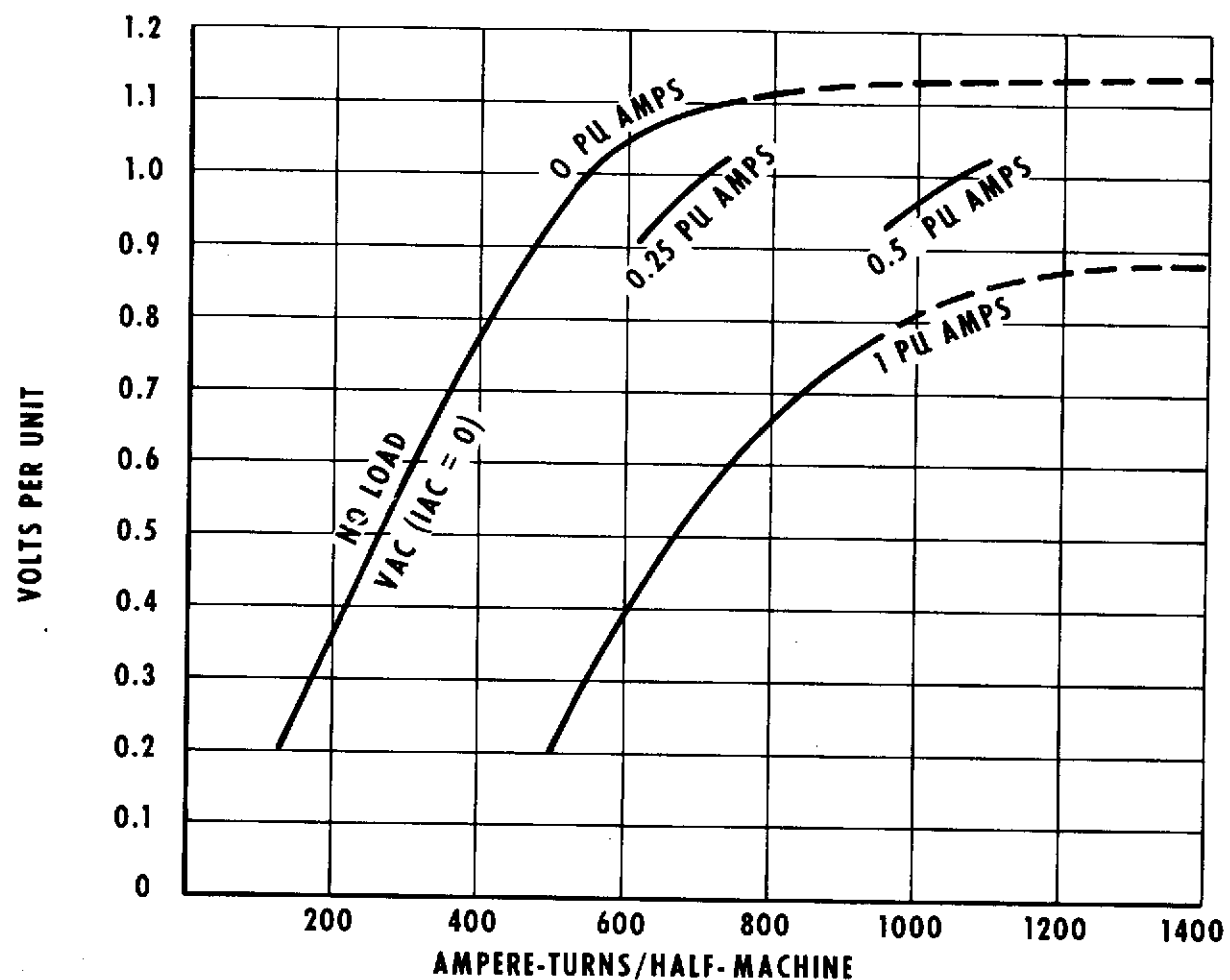


FIGURE 3-18



3.3.3 Motor Start Characteristics

Figures 3-19 and 3-20 show the expected performance characteristics of the alternator when operated as an induction motor.

For purposes of evaluation, the breakaway torque is estimated to be 0.22 to 0.33 Newton-meters (2.0 to 3.0 in.-lb); however, the maximum drag torque is expected to be about 0.33 to 0.45 Newton-meters (3.0 to 4.0 in.-lb) at about 4000 rpm. Using the torque/slip relationship of

$$\tau = \tau_{LR} s^{1/2},$$

the required stall torque to accelerate past the maximum drag torque speed is 1.58 per unit.

3.3.4 Magnetic Unbalance Characteristics

The standstill forces that would be developed with the rotor magnetically displaced and the unit excited for rated load and short circuit conditions are given as negative spring rates in Table 3-14. These forces were calculated using standard magnetic theory.

It has been demonstrated that the force is significantly reduced when the rotor is rotated. As reported in NASA CR-1452, the force is reduced by a factor of four in a homopolar inductor alternator. Tests with the Brayton Rotating Unit (BRU) four-pole Rice alternator demonstrated significant reduction, in that excitation at standstill locked the rotor; whereas the same excitation at a few hundred rpm only slightly affected the rotor orbit. A mathematical model of the Rice machine was developed to evaluate the effects of rotation on unbalance forces in the auxiliary gap. The results of the analysis indicated that the force was reduced by a factor of two, maximum, at speeds greater than 7000 rpm. A similar reduction in main gap force is expected.



MOTOR START CHARACTERISTICS I

PER UNIT RATING

65.9	VOLTS, L-N
12.27	AMPS PER PHASE
2.0	KW
2.43	KVA
3.25	IN.-LB (0.367 NEWTON-METERS)
52,000	RPM
1733	Hz
0.038	VOLTS/Hz

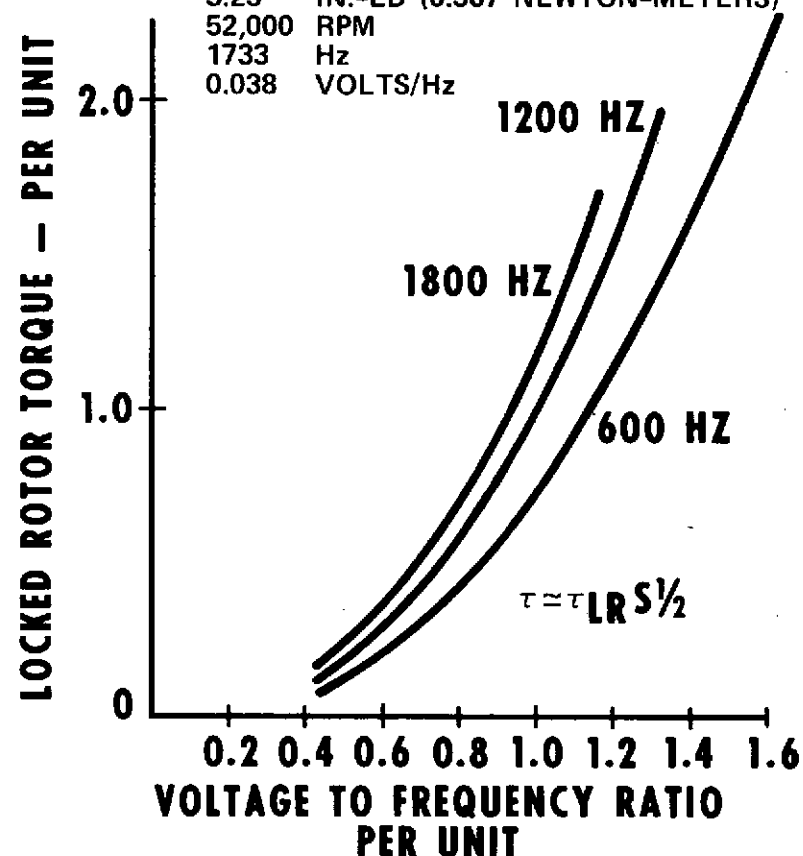
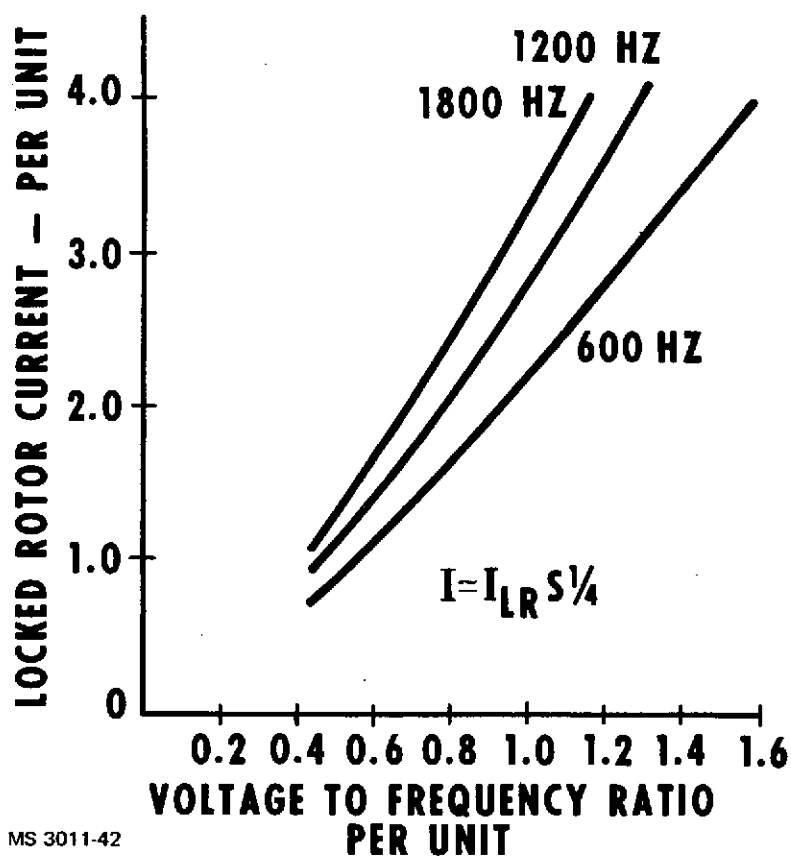


FIGURE 3-19



MOTOR START CHARACTERISTICS II

PER UNIT RATING

65.9 VOLTS, L-N
12.27 AMPS PER PHASE
2.0 KW
2.43 KVA
3.25 IN.-LB (0.367 NEWTON-METERS)
52,000 RPM
1733 Hz
0.038 VOLTS/Hz

--- KVA
— KW

ASSUMES

ROTOR WT = 2.0 LBS

ROTOR IP = 0.0022 IN-LB-SEC²

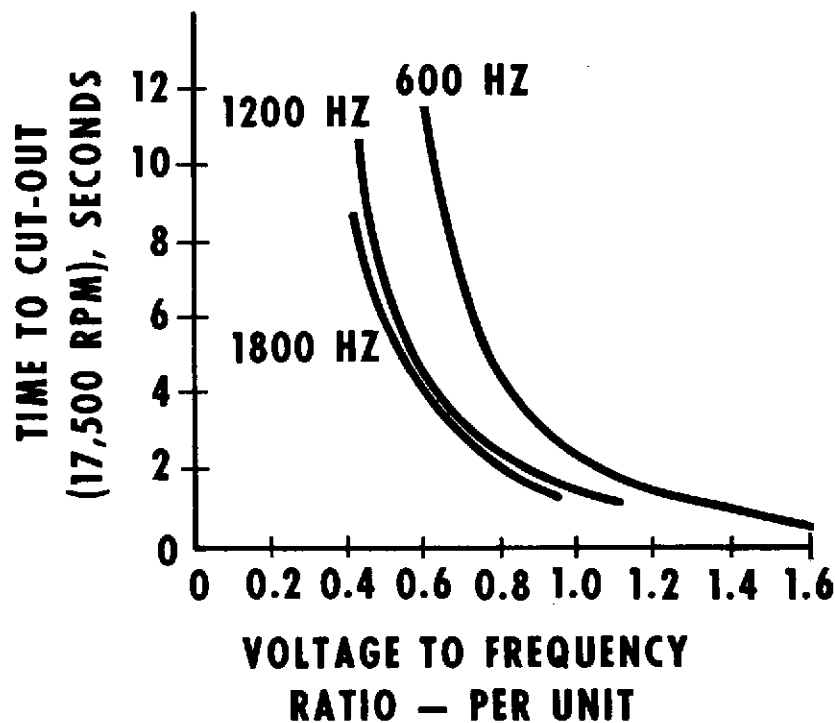
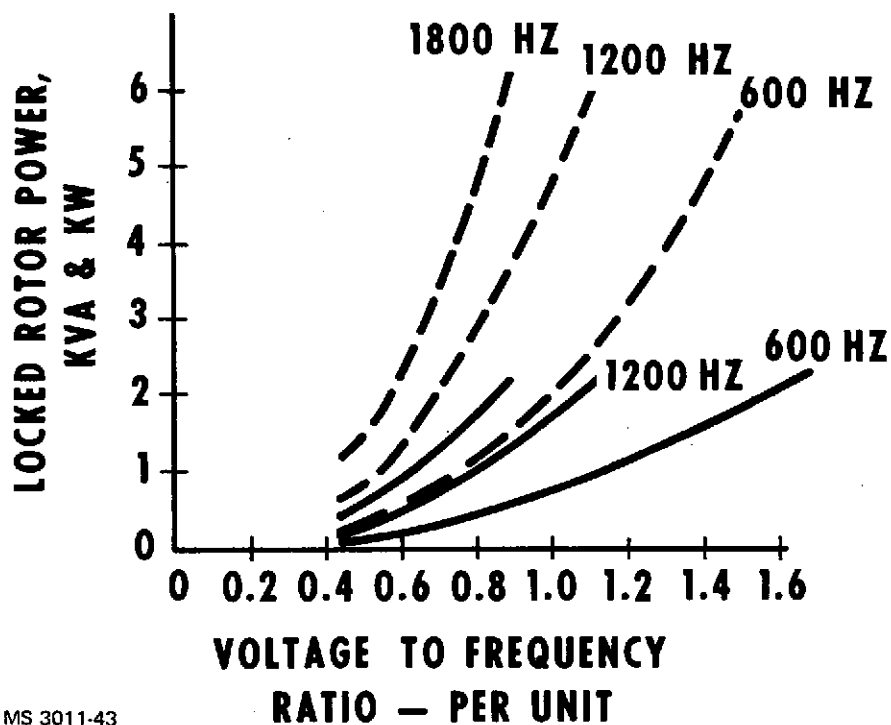


FIGURE 3-20



MAGNETIC UNBALANCE — ALTERNATOR OPERATION

STANDSTILL FORCE 'SPRING RATE'¹

LOCATION	1 CAPSULE	2 CAPSULES	3 CAPSULES	3 CAPSULES SHORT CCT (36.8 AMPS/φ)
AUXILIARY GAP (EACH BEARING) N/CM (LB/IN.) OF DISPLACEMENT	1424 (813)	2033 (1161)	2702 (1543)	2492 (1423)
MAIN GAP N/CM (LB/IN.) OF DISPLACEMENT	2988 (1706)	3710 (2118)	4360 (2490)	1470 (840)

¹VALUES ARE IN THE DIRECTION OF MINIMUM AIR GAP.

EFFECT OF ROTATION: THE SPRING RATE WILL BE REDUCED BY A FACTOR OF 2
AT SPEEDS GREATER THAN 7000 RPM AND WILL BE DIRECTED
AT AN ANGLE OF 24.5 DEGREES PRECEEDING THE MINIMUM
AIR GAP.



The effect of magnetic unbalance due to field excitation is minimized by inhibiting excitation until the machine has passed through the critical speeds and the gas bearing film is established.

Displacement will also result in magnetic unbalance forces during motor starts. Figure 3-21 shows the expected "negative spring rate" as a function of speed for a motor start with 0.003 in. displacement and using a fixed 31 v, 600 Hz supply as required to provide worst case starting torque (see Section 3.3.2).



MAGNETIC UNBALANCE — MOTOR START OPERATION

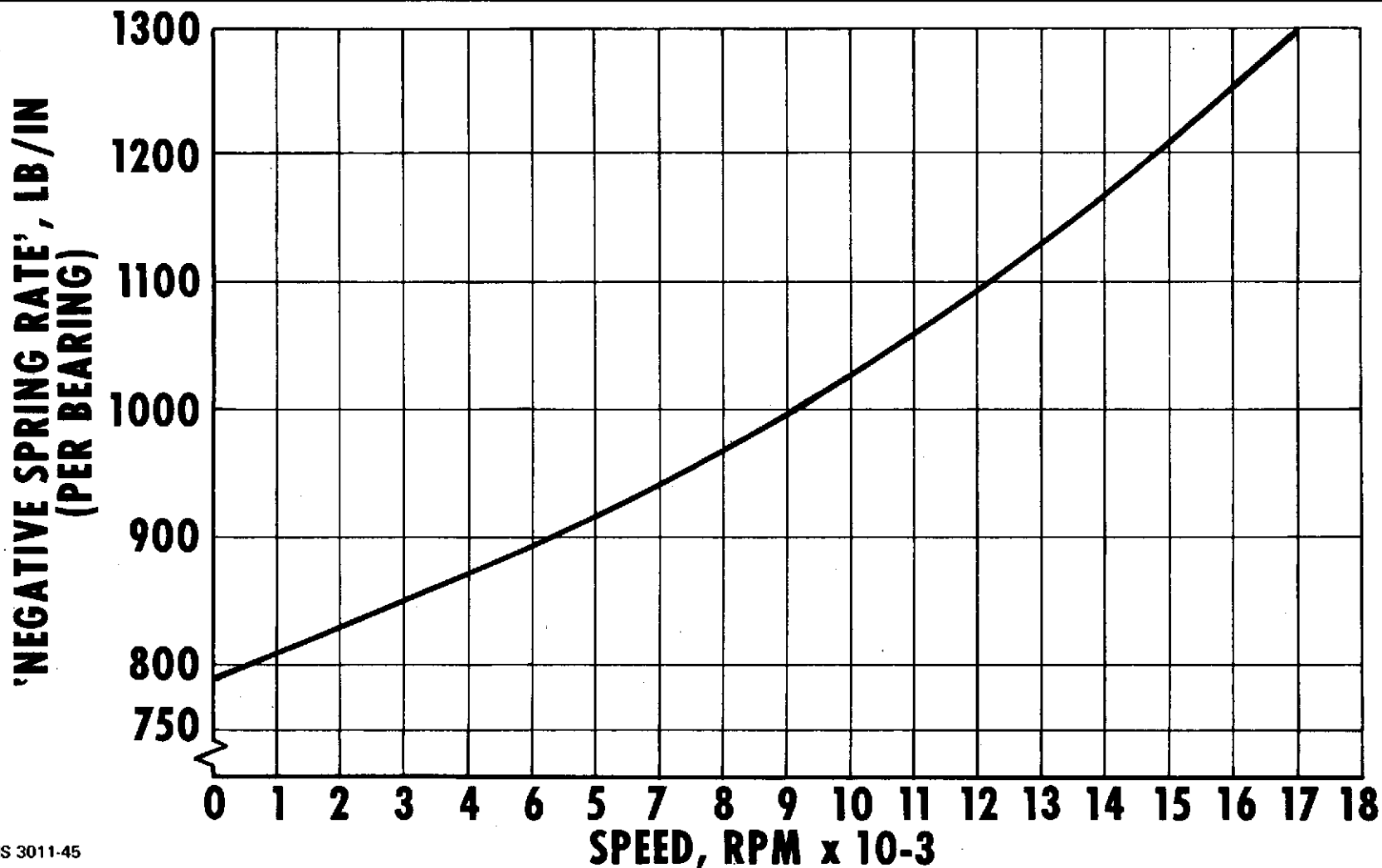


FIGURE 3-21



3.4 Thermal Analysis

3.4.1 General Approach

The rotor thermal analysis was performed in three parts. The turbine, and compressor and thrust runner were both analyzed at AiResearch, Phoenix, while the alternator portion of the rotating group was analyzed by the Los Angeles Division. This approach is shown diagrammatically in Figure 3-22.

The gas path boundary conditions for the Phoenix portion of the analysis were those specified for the three capsule reference design, shown in Figure 1-6. Several heat flexes were specified by Phoenix at the interface with the alternator (see Figures 3-23 and 3-24). Los Angeles then used these heat flux and interface temperatures as boundary conditions for thermal analysis of the alternator. The results of the combined analysis for the 3 and 1 capsule power levels are presented in Figures 3-25 and 3-26.

3.4.2 Turbine-End

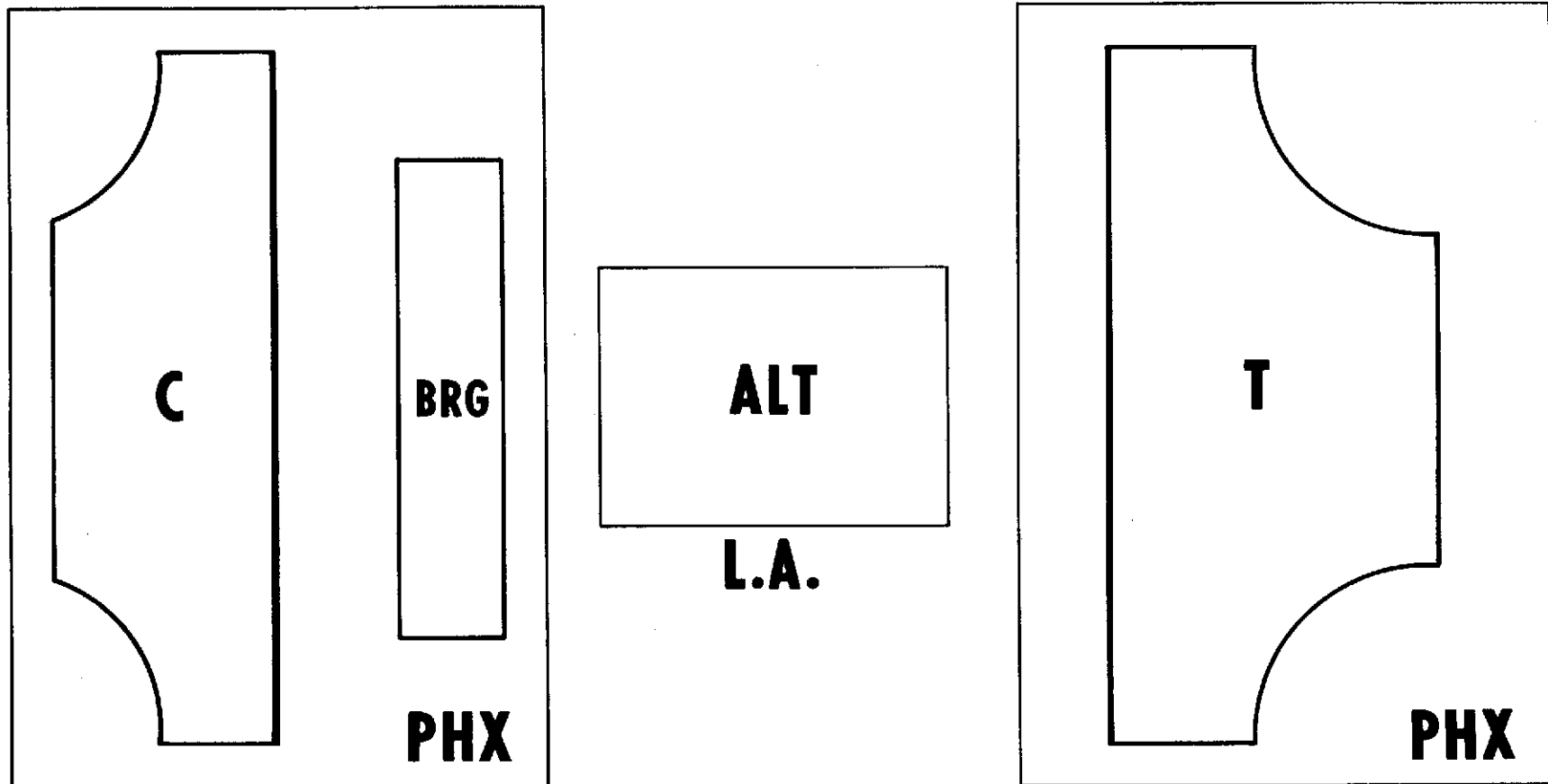
The results of the turbine-end thermal analysis for both 1 and 2 percent cooling flows are presented in Figure 3-23. The maximum equivalent stress is less than 15 ksi which, for the 713LC turbine material, operating at metal temperatures less than 1300°F, represent a very conservative design.

3.4.3 Compressor-End

The compressor and thrust runner thermal analysis data are presented in Figure 3-24. The maximum heat flow rate to the compressor of approximately 120 watts is quite low compared to that of other turbomachines designed by AiResearch, and no thermal stress problems are anticipated over the design life goal of 5 yr. In addition, the radial



THERMAL ANALYSIS PROBLEM APPROACH

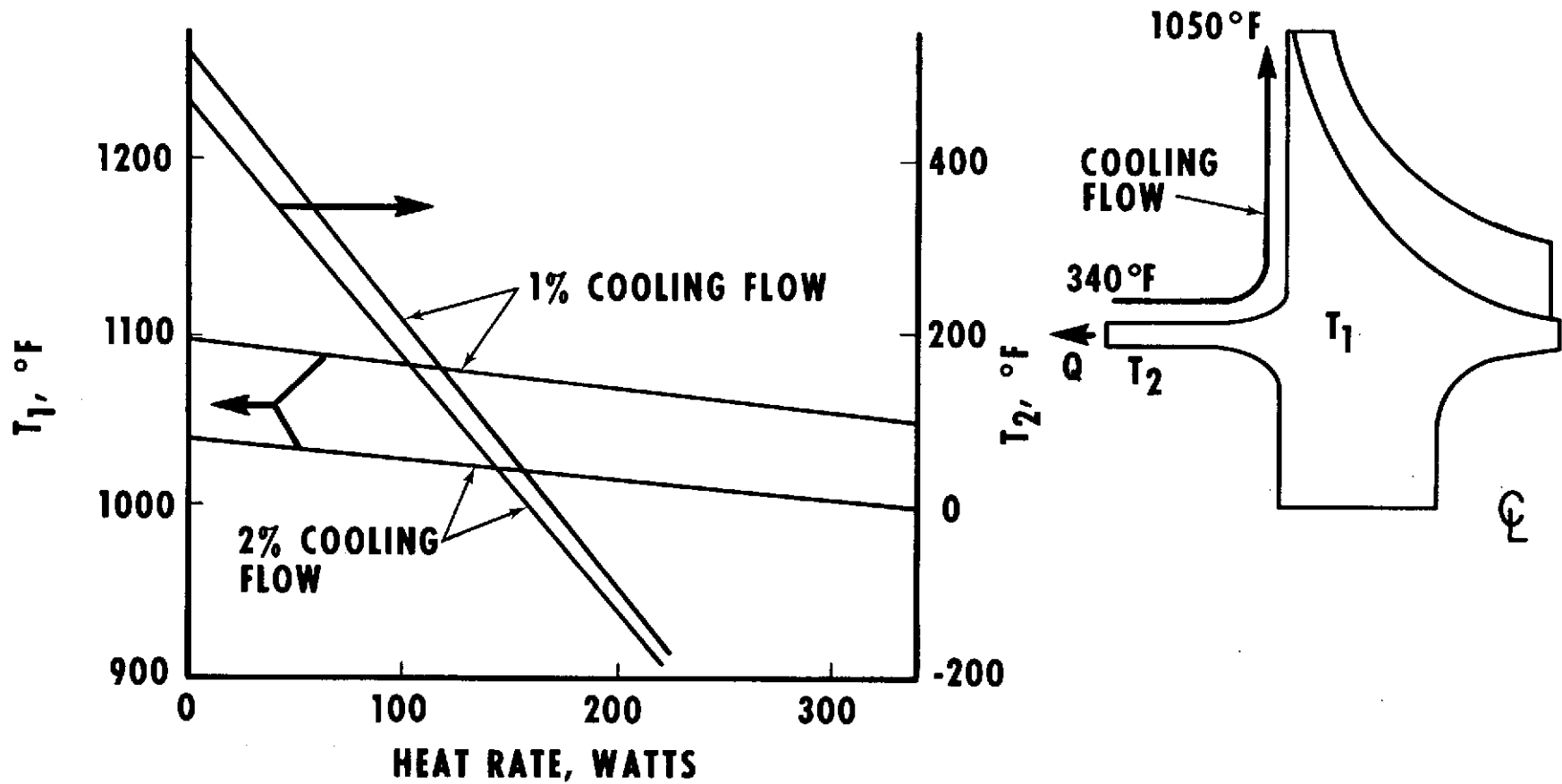


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FIGURE 3-22



TURBINE WHEEL THERMAL ANALYSIS



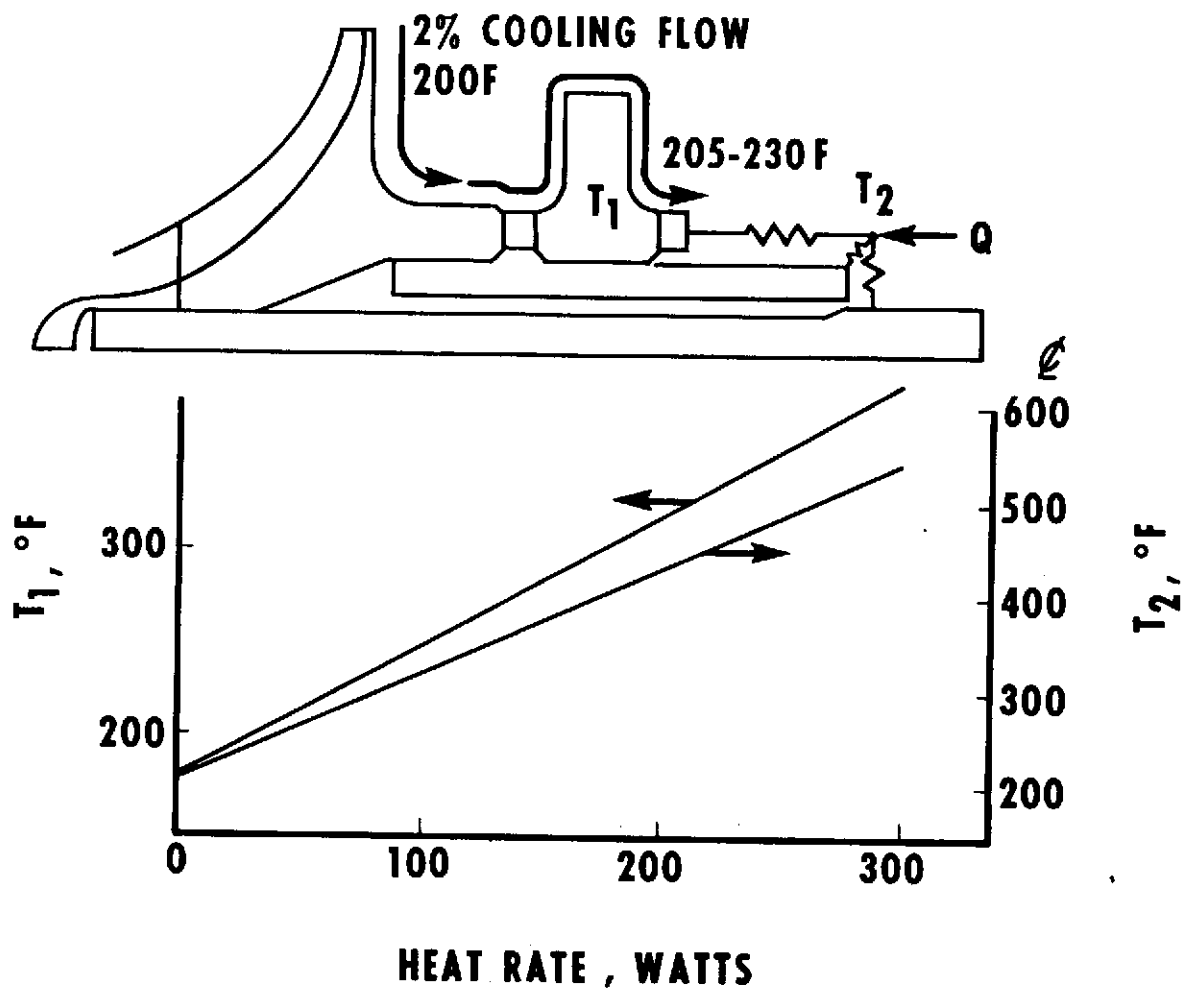
APS-5440-R
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FIGURE 3-23



COMPRESSOR AND THRUST RUNNER THERMAL ANALYSIS



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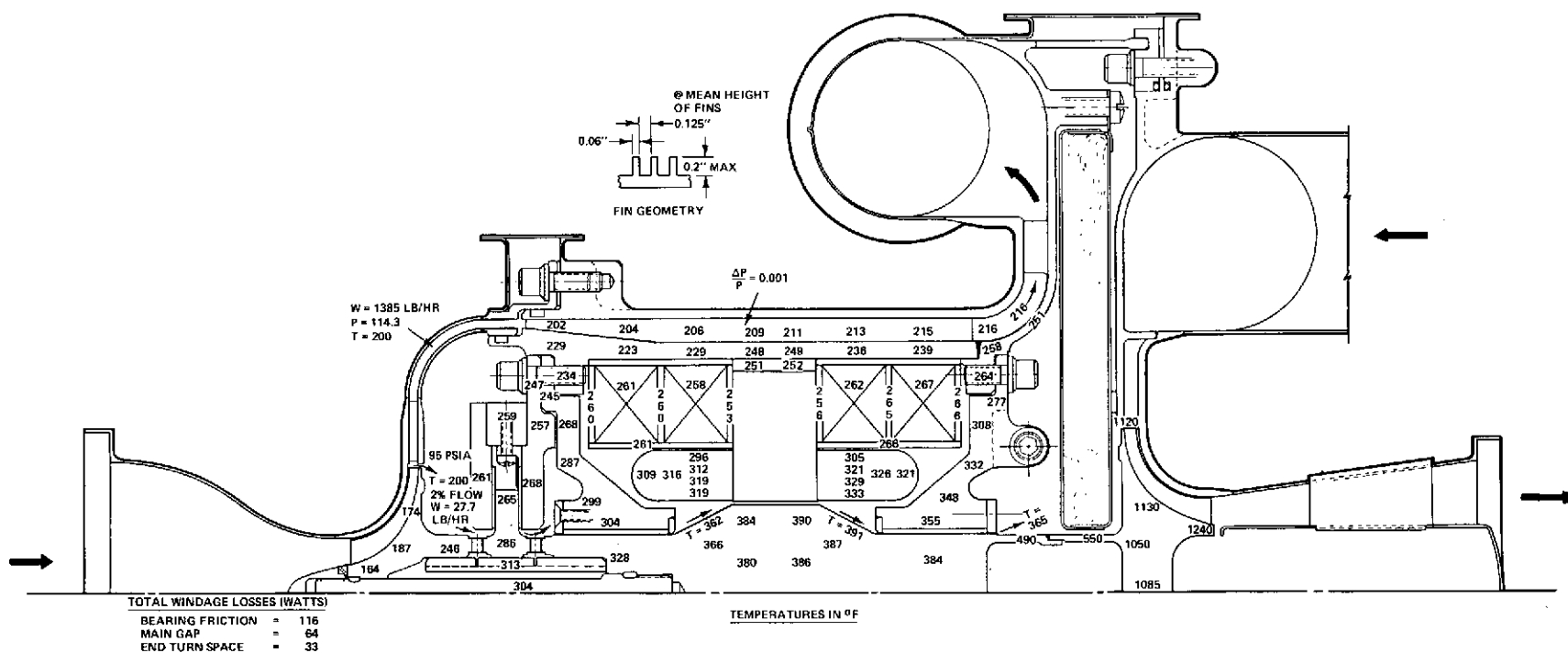
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FIGURE 3-24



THERMAL DESIGN SUMMARY — 3 CAPSULES

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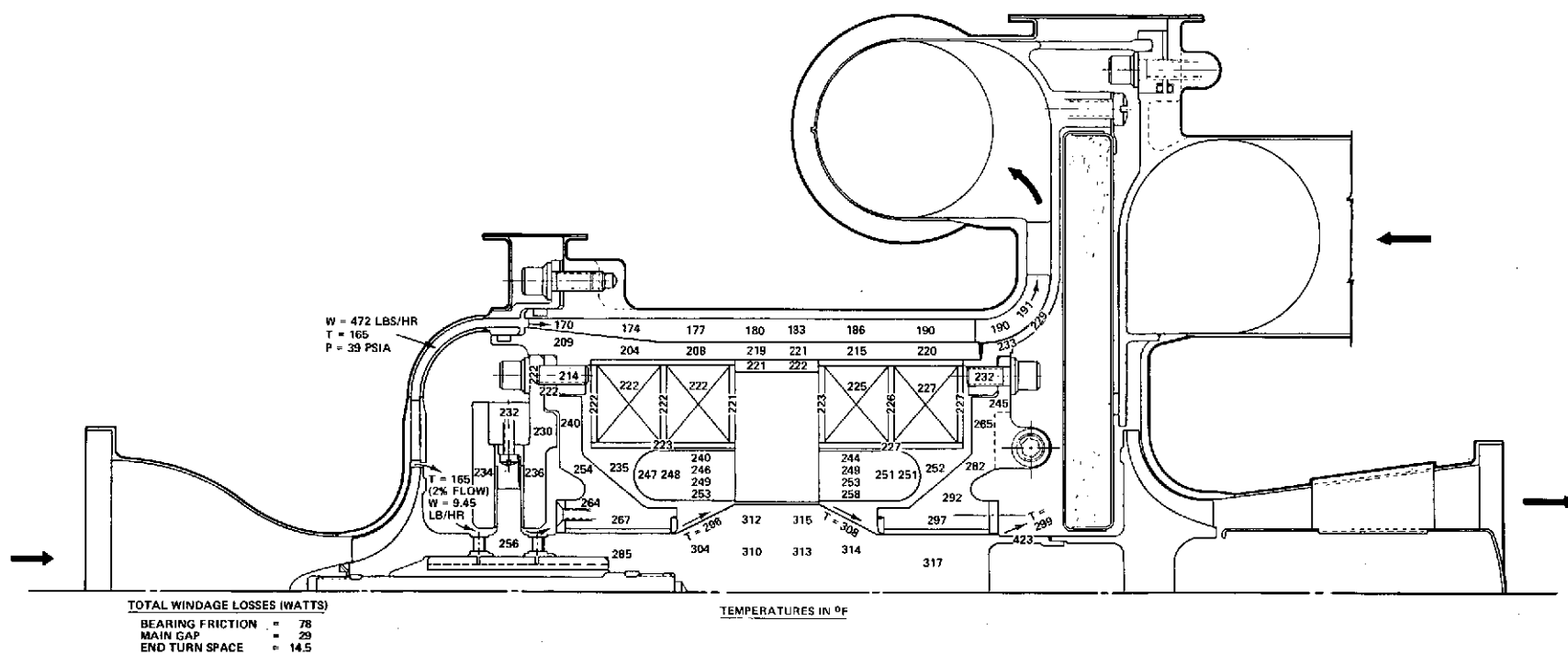
MS 3011-15

FIGURE 3-25



THERMAL DESIGN SUMMARY – 1 CAPSULE

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MS 3011-16

FIGURE 3-26



thermal gradient in the thrust rotor is small and the axial gradient even smaller. As a result, there is virtually no thermal distortion of the thrust rotor, and thrust bearing performance is unimpaired.

3.4.4 Alternator

The results of the thermal analysis on the final configuration of the mini-BRU alternator for 3 and 1 heat source capsules are presented in Figures 3-25 and 3-26. The heat load from the turbine was input as a heat flux (watts) versus shaft temperature. Heat generation in the foil journal and thrust bearings was also input to the analysis. A leakage flow of 2 percent was assumed to flow along the shaft from the compressor discharge to the back face of the turbine wheel. Temperatures in critical areas of the stator are seen to be well below the 400°F maximum tolerated and other temperatures throughout the machine are compatible with the materials used.

From the results of this analysis and other preliminary work, it appears that the final design selected was the most logical choice of all the schemes originally contemplated. The simple, machined rectangular fin approach employed in the final design (fin details on Figure 3-25) proved adequate without resorting to the complexity of a brazed plate fin design. The pressure drop in the rectangular fin design ($0.001 \Delta P/P$) was well below the design objective ($0.005 \Delta P/P$). The final design chosen improves cycle performance by adding heat after the compressor.



3.5 Heat Exchangers

3.5.1 Recuperator

During the course of the system optimization study, a number of alternative recuperator core configurations were investigated. The final selection was determined only after a detailed analysis, which included manufacturability, unit life expectancy, and performance considerations, was made.

Table 3-15 presents the reference recuperator design summary. The effectiveness of 0.975 and the fractional pressure drop of 0.70 percent (total both sides) represent the optimum combination with regard to minimizing total system weight.

3.5.2 Radiator

Many radiator configurations were investigated, and the configuration selected for the reference system is shown in Figure 3-27. The first configuration considered was steel tubes with internal fins, steel headers and ducts, and aluminum external fins and armor. Cases were run with 0, 4, 8, 16 and 32 radial fins within each tube. The bare tube case (no internal fins) showed no weight penalty over its finned counterparts. Radiators of all aluminum configuration were also investigated and presented an alternative 25 percent reduction in radiator weight, however a bimetal joining problem exists with the stainless steel ducting at two locations in the loop.

The design values for pressure drop allocation were optimized for the minimum total weight of the radiator and also for the minimum total of recuperator and radiator weight.

A summary of geometry and performance for the reference design is presented in Section 1.3.2 of this report. Throughout the analysis of the radiator, a 20 percent pressure drop margin was reserved.



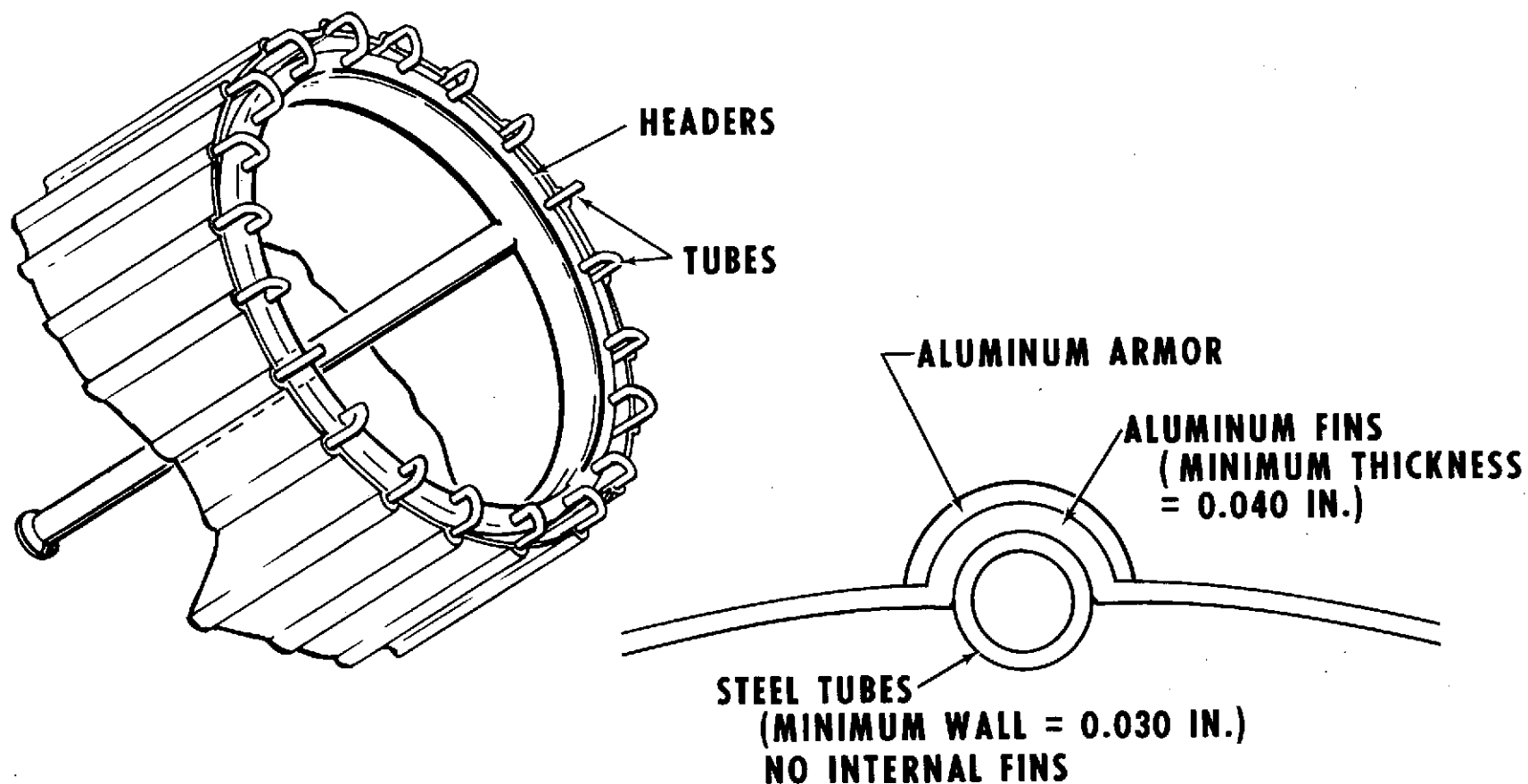
RECUPERATOR DESIGN SUMMARY

	ORIGINAL DESIGN*	NEW DESIGN*
FIN CONFIGURATION: NUMBER OF FINS PER INCH FIN HEIGHT (INCHES) FIN THICKNESS (INCHES) FIN MATERIAL	28. 0.154 0.004 NICKEL	20. 0.101 0.004 HAST-X
PLATE CONFIGURATION: PLATE THICKNESS (INCHES) PLATE MATERIAL	0.006 HAST-X	0.008 HAST-X
HEADER CONFIGURATION: NUMBER OF FINS PER INCH FIN THICKNESS (INCHES) FIN MATERIAL	10. 0.006 NICKEL	10. 0.006 HAST-X
GENERAL: SPACER BAR THICKNESS (INCHES) WRAP-UP THICKNESS (INCHES) BRAZE THICKNESS (INCHES)	0. 0.060 0.	0.100 0.060 0.001

*MATRIX FOR BOTH HIGH AND LOW PRESSURE SIDES.



RADIATOR MODEL



MS 3063-24

FIGURE 3-27